WL-TR-97-3081

CONCEPTUAL DEVELOPMENT OF AN ADVANCED CYCLE BARE BASE ENVIRONMENTAL CONTROL UNIT

VOLUME 2 - ADDITIONAL INFORMATION

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NOVEMBER 1995

FINAL DRAFT REPORT FOR 10/01/93 - 10/31/95

FLIGHT DYNAMICS DIRECTORATE
WRIGHT LABORATORY
AIR FORCE MATERIEL COMMAND
TYNDALL AFB FL 32403-5323

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REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY (*Leave blank*)

2. REPORT DATE
November 1995

3. REPORT TYPE AND DATES COVERED
FINAL 10/01/93 - 10/31/95

4. TITLE AND SUBTITLE
CONCEPTUAL DEVELOPMENT OF AN ADVANCED CYCLE BARE BASE ENVIRONMENTAL CONTROL UNIT
VOL 2 - ADDITIONAL INFORMATION

5. FUNDING NUMBERS
C: F08635-93-C-0018
PE: 6.2

6. AUTHOR(S)
W. GRAEBEL, C. MOIOLA, R. BOEHM

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
NEVADA ENGINEERING RESEARCH & DEVELOPMENT SYSTEMS
6452 EAST VIEWPOINT DRIVE
LAS VEGAS, NV 89115

8. PERFORMING ORGANIZATION REPORT NUMBER

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
FLIGHT DYNAMICS DIRECTORATE
WRIGHT LABORATORY
AIR FORCE MATERIEL COMMAND
TYNDALL AFB FL 32403-5323
POC: Kevin Grosskopf, WL/FIVCF, 904-283-3732

10. SPONSORING/MONITORING AGENCY REPORT NUMBER
WL-TR-97-3081

11. SUPPLEMENTARY NOTES
Also see WL-TR-97-3080

12a. DISTRIBUTION/AVAILABILITY STATEMENT
APPROVED FOR PUBLIC RELEASE
DISTRIBUTION UNLIMITED

12b. DISTRIBUTION CODE

13. ABSTRACT (*Maximum 200 words*)

The objective of the present concept study was to critically review and assess innovative advanced technology concepts of heating and cooling which can be applied to ECUs to support the Global Reach-Global Power strategy of today's USAF mission while protecting the environment. Nine innovative technologies were reviewed and of these, six were rejected as unsuitable to Air Force ECU needs at the present time. Of the remaining three, Stirling cycle heat pump technology was selected as the most suited to Air Force advanced needs. Hybrid Stirling systems such as the "pulse-tube" have been identified as being capable to meet all major performance criteria such as appreciable reductions in both weight and volume, increased energy efficiency, and the elimination of occupationally hazardous or environmentally suspect refrigerants. Gains in these areas within follow-on R&D efforts will provide forward deployed forces a more capable, less logistically taxing ECU system.

14. SUBJECT TERMS
AIR FORCE, ENVIRONMNETAL CONTROL UNIT (ECU), ENERGY EFFICIENCY, POWER FACTOR, OZONE DEPLETING CHEMICAL, HEAT TRANSFER

15. NUMBER OF PAGES
147

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
UNCLASSIFIED	UNCLASSIFIED	UNCLASSIFIED	SAR

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std Z39-18
298-102

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PREFACE

This report was prepared by Nevada Engineering Research & Development Systems, 6452 East Viewpoint Drive Las Vegas, Nevada 89115-7052 under contract F08635-93-C-0018, Tyndall AFB, FL 32403-5323.

The report summarizes work performed between 24 September, 1993 and 24 September 1995. Mr Kevin R. Grosskopf was the WL/FIVCF Project Manager.

A. Acoustic Heat Pumps

A.1 General Description.

Acoustic heat pumps consist of a closed, gas-filled tube with a piston driver or acoustic electrodynamic speaker mounted at one end. The driver sets up a standing monochromatic acoustic wave in the tube. Figure A.1 is a schematic drawing showing the placement of the driver, heat exchangers, and internal plates in a one-half wave length tube. An Helmholtz resonator can be used to replace a portion of the length of the tube, thereby making the tube length shorter but the overall device bulkier. Reducing the length also decreases losses in the tube somewhat. The relation of the length of the tube to the acoustic wave length must be carefully maintained for successful heat pumping. This is accomplished by monitoring of the acoustic pressure wave and adjusting the driver frequency to match current acoustic conditions.

Measuring from a pressure node, the acoustic velocity (v_s), pressure (p_s), and temperature (T_s) in the tube are given in terms of the driver frequency (f), position (x), sonic speed (c_a) and time (t) by

$$v_{total} = v_s = V_1 \cos(2\pi ft) \cos(2\pi fx/c_a), \qquad (A.1)$$

$$p_{total} = p_a + p_s,\ p_s = P_1 \sin(2\pi ft) \sin(2\pi fx/c_a), \qquad (A.2)$$

$$T = T_a + T_s,\ T_s = T_1 \sin(2\pi ft) \sin(2\pi fx/c_a), \qquad (A.3)$$

where the subscript "a" denotes ambient (static) conditions in the tube, the subscript "s" refers to the standing acoustic waves, and the subscript "1" refers to the amplitudes of the acoustic quantities. The amplitudes of the standing acoustic pressure (P_1) and temperature (T_1) waves are related to the amplitude of the velocity wave (V_1) by

$$P_1 = \rho_a c_a V_1,\ T_1 = (\gamma - 1)T_a P_1/\gamma p_a = (\gamma - 1)T_a \rho_a c_a V_1/\gamma p_a, \qquad (A.4)$$

where ρ_a is the ambient density of the gas and γ is the ratio of the specific heats.

The quantity $Z = p_s/Av_a$, with A denoting the cross-sectional area of the tube, is called the local acoustic impedance of the gas. It is the pressure needed to provide a unit volume change. From equations (A.1), (A.2), and (A.4),

$$Z = \rho_s c_a \tan(2\pi fx/c_a)/A. \qquad (A.5)$$

The piston driver (speaker) is placed in a tube at a point where its acoustic impedance matches the acoustic impedance of the tube.

The rate of work per unit volume of gas produced in the gas by this standing acoustic wave is given by

$$W = \int p_s v_s \, dt, \quad \text{(A.6)}$$

the integration being performed over one period (1/f). If no solid objects are placed within the tube, the net work done over a period is zero since the acoustic pressure and velocity are 90 degrees out of phase.

To obtain useful work from this device, an array of stacked plates is inserted into the tube near one end, thereby establishing viscous and thermal boundary layers on each of the plates. The thicknesses of the velocity and thermal boundary layers are given by

$$\delta_v = \sqrt{\mu/\pi f \rho_a} \text{ and } \delta_t = \sqrt{k/\pi f \rho_a c_p}, \quad \text{(A.7)}$$

where k is the thermal conductivity of the gas, μ is the viscosity of the gas, and c_p is the specific heat of the gas at constant pressure. The square of the ratio of these two boundary layer thicknesses is called the Prandtl number, viz.

$$Pr = (\delta_v/\delta_t)^2 = c_p/\mu k, \quad \text{(A.8)}$$

which is solely a function of the physical properties of the gas. It is an important parameter in the performance of acoustic heat pumps, as making the Prandtl number low by a proper choice of the working fluid minimizes losses.

The velocity boundary layer is analogous to one first studied by G. Stokes over a hundred years ago. It is discussed in many advanced fluid mechanics text books and is frequently referred to as Stokes second problem. In the Stokes solution the plate is oscillated periodically and the fluid is otherwise at rest. In the thermoacoustic problem the gas is oscillating while the plate is at rest. The pressure field in both cases is unaffected by the presence of the plate, the flow being due solely either to the motion of the plate or of the outer fluid. In both cases the velocity field is altered by the effect of viscosity, to the extent that in regions of the boundary layer the flow is actually 180 degrees out of phase with the driving cause of the flow. This phase difference allows a positive amount of work to be done by the gas, and a temperature gradient is established along the tube. The result is that heat is transferred along the plate towards a pressure antinode.

For a single plate the acoustic velocities and temperature in the boundary layers can be found to be

$$v_{bl} = V_1[\cos(2\pi ft) - \exp(-y/\delta_v)\cos(2\pi ft - y/\delta_v)]\cos(2\pi fx/c_a), \quad \text{(A.9)}$$

$$T_{bl} = T_1[\sin(2\pi ft) - \exp(-y/\delta_t)\sin(2\pi ft - y/\delta_t)]\sin(2\pi fx/c_a)$$

$$+ T_p(x)\exp(-y/\delta_t)\cos(2\pi ft - y/\delta_t), \quad \text{(A.10)}$$

where T_p is the temperature of the plate and y is the distance measured perpendicular to the plate. It is seen from the terms expressed as an exponential times a trigonometric term that there are regions where the sign of the velocity is reversed compared to the flow outside of the boundary layer. Since pressure is the same inside and outside of the boundary layer, these regions of phase reversal are associated with positive work being done. This phenomenon is referred to as "acoustic streaming".

The analysis of an acoustic heat pump is quite complicated, and cannot be given in a simple closed form. The basic theory was established by Rott {Rott 1969, 1973, 1974a and b, 1976, 1980,1984a and b], and the specifics for a tube containing a family of plates has been elaborated by a number of investigators [Merkli and Thomann 1975a and b, Wheatley, Hoffler et al 1983a and b, Wheatley 1983 and 1984, Wheatley, Hoffler et al 1985, Wheatley, Swift, and Migliori 1986, Atchley, Hoffler et al 1990]. When viscous effects are small, a reasonable approximation for the efficiency is given by

$$\eta = \eta_{carnot}/\Gamma, \tag{A.11}$$

while for refrigeration a similar approximation gives for the COP

$$COP = \Gamma \cdot COP_{carnot}. \tag{A.12}$$

The dimensionless parameter Γ is the ratio of two temperature gradients

$$\Gamma = \nabla T_p / \nabla T_{crit}, \tag{A.13}$$

where ∇T_p is the temperature gradient induced along the plates. For a plate of length L, ∇T_{crit} is given by

$$\nabla T_{crit} = \frac{(\gamma - 1)}{\beta L} \tan(x/L) \approx \frac{T_p}{L}, \tag{A.14}$$

where γ is the ratio of the specific heats and β is the thermal expansion coefficient of the gas.

Analytical determination of the temperature on the plate is a complicated task. Simply put, the addition of plates within the tube allows a temperature variation throughout the boundary layers on the plates which results in a heat flux parallel to the plates. The phase shifts induced in acoustic pressure and velocity allows the generation or absorption of acoustic power by the fluid near the plate, thus turning the acoustic tube into a heat engine.

The actual expressions for heat flux and acoustic power are quite complicated. The form for low Prandtl number fluids with $\delta_v/y_o \sim \sqrt{Pr}$ and $y_o \sim \delta_t$ give qualitative results sufficiently accurate for physical interpretation. The total heat flux produced along the plate is given by

$$\dot{Q} = -\Pi[\tfrac{1}{4}\delta_k T_m \beta p_1^s \langle v_1^s \rangle (\Gamma - 1)/(1+\epsilon_s)(1-\sqrt{Pr})$$

$$+ (y_o k + \ell k_s) dT_p/dx] \qquad \text{(A.15)}$$

and the total acoustic power produced along the plate is given by

$$\dot{W} = \tfrac{1}{4}\omega\Pi\, \Delta x[\delta_k(\gamma-1)(p_1^s)^2(\Gamma - 1)/\rho_a c_a^2(1+\epsilon_s) - \delta_v \rho_a \langle v_s \rangle^2], \qquad \text{(A.16)}$$

where y_o is the spacing between plates and ℓ is the plate thickness. In these expressions Π is the width and Δx the length of the inserted plates. The quantities p_1^s and v_1^s are the amplitudes of the standing pressure and velocity waves in the boundary layers. The quantity $\langle v_1^s \rangle$ is the average of the acoustic velocity over the half-distance between plates. The parameter ϵ_s is given by

$$\epsilon_s = \sqrt{\rho_a c_p \delta_k}\tanh[(1 + i)y_o/\delta_k]/\sqrt{\rho_s c_s \delta_s}\tanh[(1+ i)\ell/\delta_s]. \qquad \text{(A.17)}$$

The quantities ρ_s and c_s are the density and specific heat of the plate material, and

$$\delta_s = \sqrt{k_s/\pi f \rho_s c_s.} \qquad \text{(A.18)}$$

is the thermal penetration depth in the plate.

The second terms in the expression for the heat flux and acoustic power rates are the effects of viscous dissipation in the boundary layer. When the Prandtl number is small, these terms are minimized. Fluids which have small Prandtl numbers are the liquid metals with Prandtl numbers around 0.01, superfluid helium mixtures with Prandtl numbers near 0.1, and binary monatomic gas mixtures with Prandtl numbers near 0.3.

The efficiency and COP of the acoustic heat pump are at their maximum values when $\Gamma = 1$. However, it is seen that when $\Gamma = 1$, the acoustic power at this condition vanishes except for viscous effects, as does the heat flux. The fact that both the heat flux and acoustic power generated are proportional to the square of the amplitudes of the standing waves indicates that the thermoacoustic effect is second order in the acoustic power introduced into the tube by the speaker.

The efficiency and COP described above do not include the power losses which occur in transforming electric power to acoustic power by means of a loudspeaker or other acoustic driver. Typical speaker efficiencies are in the 3% to 50% range.

Designs for thermoacoustic refrigerators have been presented by a number of investigators [Bolos 1990, Garrett 1991, Garrett and Hoffler 1992]. However the detailed theory needed for assessing the direction for improved designs is still lacking. The above analysis assumes that the plate length is short compared to the wavelength of the standing wave, that ∇T_p is small compared to the mean temperature, and that the plate does not affect the mean acoustic field. The latter assumption is particularly suspect. The theory does point out that the Prandtl number should be kept low to reduce viscous effects and that $T_m\beta$ should be large. The plate should ideally have a low thermal conductivity along the plate and a high thermal conductivity perpendicular to the plate. More experiments and theory are needed to improve on the design criteria.

A.2 Advantages.

a. No refrigerant is needed other than the gas in the tube. Air or water may be satisfactory for heat transfer external to the heat pump.

b. Changing from cooling to heating requires only a change in air ducting.

c. These heat pumps are compact in size. The units can be stacked so as to be in parallel thermally.

A.3 Disadvantages.

a. Transformation of electrical energy to acoustic energy via a loudspeaker usually results in a large loss of energy, decreasing the overall efficiency and COP of the system.

b. Output heating/cooling power per unit volume is relatively small.

c. Applications to date have currently been for small cooling capacities, either in space applications or in cooling instruments in deep wells.

d. Substantial noise is generated by the driving speaker. For an open tube sound levels of the order of 100 db have been measured at a distance of 1 meter. These sounds are at frequencies which are well within the audible range (about 500 hz).

e. To achieve results it is necessary to use gases with low Prandtl numbers such as helium. Pressures used to date have been in the 10 bar range (150 psi). These pressures are needed to obtain satisfactory gas densities and acoustic velocities. Helium is a difficult gas to contain within a tube, as its low atomic weight means that it can pass through normal metal walls unless expensive plating is applied to the interior of the tube. Long-term sealing of joints, particularly in environments where they may be subject to strong vibrations, can also be expected to be a problem. A make-up gas source is a necessity.

f. The driver must be acoustically well tuned to the impedance of the system to lower power losses. An electronic feedback system is necessary to monitor and change the driving frequency to match the changes in the acoustic system during operation. Magnetic suspension of the driver has been found to be desirable.

g. Heat energy must be brought into and out of the tube by means of very small heat exchangers which must be designed for high efficiency. The diameter of a typical acoustic tube lies between one and two inches in order to keep cross-harmonics from developing.

The heat exchanger would have to be in that size range, and would have to be thin enough so that it did not interfere or change the acoustic waves. Liquid heat transfer loops are necessary to connect the heat exchangers to ensure adequate heat transfer, but the necessity for non-interference with the acoustic waves means that only thin plates of copper or another good thermal conductor could be used inside the tube as a heat exchange medium.

A.4 Tradeoffs, Hazards, and Risks Analysis.

Hydrogen and helium are desirable gases from a thermodynamic standpoint for these heat pumps, but they pose difficulties because of leakage, requiring a make-up procedure for the gas. Also, hydrogen is highly combustible. The leakage problem would pose a severe storage problem. Use of air as the gas medium is a possibility, but its higher Prandtl number (0.5 to 0.8, depending on the temperature) lowers the heat pumping effectiveness. The elevated pressures necessary to achieve desirable acousto-thermal properties also poses a hazard potential to operators and passers-by. The control system would likely be fairly complex, having to control each tube separately as small individual changes in temperature and pressure in each tube must be accounted for. Because of the large number of tubes needed to develop sufficient cooling power, the devices would be bulky.

A.5 Manufacturers.

None to date.

A.6 References.

Atchley, A. A., T. J. Hoffler, M. L. Muzzerall, M. D. Kite, and C. Ao, "Acoustically generated temperature gradients in short plates", J. Acoust. Soc Am. vol. **88** (1), pp. 251-263, 1990.

Bolos, J., "Technology transfer and cooperative research and development agreements", Nav. Res. Rev. vol. **42** (3), pp. 31-33, 1990.

Garrett, S. L., "Thermoacoustic life sciences refrigerator", NASA Tech. Rept. No. LS-10114. Houston TX: Johnson Space Center, Space & Life Sciences Directorate, Oct. 30, 1991.

Garrett, S. L., and T. J. Hofler, "Thermoacoustic refrigeration", ASHRAE J. Dec., pp. 28-36, 1992.

Hoffler, T. J., "Thermoacoustic refrigerator design and performance", Ph.D. dissertation, Physics Dept. U. Cal. at San Diego, 1986.

Merkli, P., and H. Thomann, "Transition to turbulence in oscillating pipe flow", J. Fluid Mech. vol. **68**, pp. 567-575, 1975.

Merkli, P., and H. Thomann, "Thermoacoustic effects in a resonant tube", J. Fluid Mech. vol. **70**, pp. 161-177, 1975.

Rott, N., "Damped and thermally driven acoustic oscillations in wide and narrow tubes", Zeit. Angew. Math. Phys. vol. **24**, p. 230, 1969.

Rott, N., "Thermally driven acoustic oscillations part II: stability limit for helium", Zeit. Angew. Math. Phys. vol. **24**, p. 54, 1973.

Rott, N., "The influence of heat conduction on acoustic streaming", Zeit. Angew. Math. Phys. vol. **25**, p. 417, 1974.

Rott, N., "The heating effect connected with non-linear oscillations in a resonance tube", Zeit. Angew. Math. Phys. vol. **25**, p. 619, 1974.

Rott, N., "Ein Rudimentarer Stirlingmotor", Neue Zuercher Ztg. vol. **197** (210), 1976.

Rott, N., "Thermoacoustics", Adv. in Applied Mech. vol. **20**, p. 135, 1980.

Rott, N., "A simple theory of the Sondhauss tube" in *Recent Advances in Aeroacoustics*, pps. 327, Krothapalli, A., and C. A. Smith, eds., Springer, New York, 1984.

Rott, N., "Thermoacoustic heating at the closed end of an oscillating gas columns", J. Fluid Mech. vol. **145**, p. 1, 1984.

Swift, G., "Thermoacoustic engines", J. Acoust. Soc. Am. vol. **84** (4), pp. 1145-1180, 1988.

Wheatley, J., T. J. Hoffler, G. W. Swift, and A. Migliori, "Experiments with an intrinsically irreversible acoustic heat engine", Phys. Rev. Lett. vol. **50**, pp. 499-502, 1983.

Wheatley, J., T. J. Hoffler, G. W. Swift, and A. Migliori, "An intrinsically irreversible thermoacoustic heat engine", J. Acoust. Soc. Am. vol. **74**, pp. 153-170, 1983.

Wheatley, J., "Acoustical heat pumping engine", U.S. Patent No. 4,398,398, Aug. 16, 1983.

Wheatley, J., "Intrinsically irreversible heat engine", U.S.Patent No. 4,489,553, Dec. 25, 1984.

Wheatley, J., T. J. Hoffler, G. W. Swift, and A. Migliori, "Understanding some simple phenomena in thermoacoustics with applications to acoustical heat engine", Am. J. Phys. vol. **78**, pp. 147-162, 1985.

Wheatley, J., G. W. Swift, and A. Migliori, "The natural heat engine", Los Alamos Science, no. **14**, 32 pages, Fall 1986.

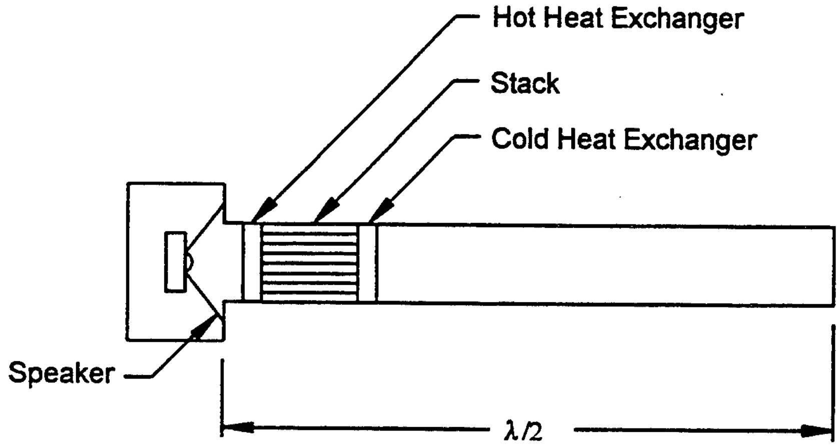

Figure A.1. Schematic of a half-wave length acoustic heat pump.

B. Brayton Cycle Heat Pumps.

B.1 General Description.

Brayton cycle heat pumps use gas as refrigerants. Usually the gas is air. In a closed cycle, the gas is confined to a closed loop, similar to the refrigerant in a vapor compression cycle. Because of the poor performance of gas-to-gas heat exchangers, frequently the cold side heat exchanger is removed and replaced by either the room or the outside air. In an open cycle, air enters the compressor from the surroundings and exits the turbine back to the surroundings; although the surroundings are not necessarily in the same space (i.e. one outside and one inside, respectively). Work produced by the turbine is fed back to the compressor to increase the efficiency of the overall system. Brayton heat pumps are used extensively in jet aircraft air conditioning because of the availability of compressed air from the engines at little cost in engine efficiency.

A typical open Brayton cycle heat pump schematic is shown in Figure B.1, with the temperature-entropy diagram given in Figure B.2. It is seen from Figure B.2 that increasing the pressure of the air also increases its temperature. This increase in temperature allows heat to be rejected though the hot heat exchanger to the high temperature reservoir during a constant pressure process. The air is then expanded through a turbine, decreasing the pressure back to atmospheric where it is rejected at a temperature lower than that of the air entering the compressor.

During the expansion and compression of the air the pressure and temperature ideally follow a polytropic process according to

$$(T_{out}/T_{in})_{expansion} = (p_{low}/p_{high})^{(1-1/\gamma)} \qquad \text{(B.1)}$$

and

$$(T_{out}/T_{in})_{compression} = (p_{high}/p_{low})^{(1-1/\gamma)}. \qquad \text{(B.2)}$$

The compressor and turbine can be described by their enthalpy changes and mechanical efficiencies according to

$$\eta_{compressor} = (h_{out,s} - h_{in})/(h_{out} - h_{in}) \qquad \text{(B.3)}$$

and

$$\eta_{turbine} = (h_{in} - h_{out})/(h_{in} - h_{out,s}) \qquad \text{(B.4)}$$

The rate of heat loss or gain in any portion of the cycle can be determined by the general equation

$$\dot{Q} = \dot{m}c_p\Delta T \qquad \text{(B.5)}$$

where ΔT is the temperature change in that portion of the cycle. The refrigeration coefficient of performance ($COP_{refrigeration}$) is expressed as

$$COP_{refrigeration} = \frac{\text{useful heat transfer}}{\text{work input}}$$

$$= \frac{Q_{evaporator}}{W_{compressor} - W_{turbine}}$$

$$= \frac{T_{room} - T_{turbine\ out}}{(T_{compressor\ out} - T_{compressor\ in}) - (T_{turbine\ in} - T_{turbine\ out})} \quad . \qquad (B.6)$$

The heating COP is similarly defined as

$$COP_{heating} = \frac{\text{useful heat transfer}}{\text{work input}}$$

$$= \frac{Q_{condenser}}{W_{compressor} - W_{turbine}}$$

$$= \frac{T_{compressor\ out} - T_{turbine\ in}}{(T_{compressor\ out} - T_{compressor\ in}) - (T_{turbine\ in} - T_{turbine\ out})} \quad . \qquad (B.7)$$

The pressure ratio p_{high}/p_{low} is the ratio of the pressure increase across the compressor. The lower pressure in an open cycle is always atmospheric because the air comes from and is returned to the surroundings. Because the heat rejection and absorption steps are constant pressure processes there are only two pressures in the cycle and thus the ratio is an important parameter describing the cycle. In an ideal analysis the pressure ratio must be specified to solve the equations. The optimum pressure ratio, which results in the highest COP, can be explicitly determined and then used to solve the equations.

Figures B.3 and B.4 are plots of computed refrigeration coefficients of performance as functions of ambient temperature and ΔT, where $\Delta T = QR$ is the temperature difference between the air in the heat exchanger and the air in the surroundings. R is the thermal resistance of the heat exchanger, accounting for losses which occur in the heat transfer process. Figure B.3 shows curves of constant ΔT, while Figure B.4 shows curves of constant ambient temperatures. When ΔT is small the COP is a modest value in the neighborhood of 0.1, increasing to values near 0.31 at ΔTs near 12°C. Compressor and turbine efficiencies of 80% were assumed in the analysis.

B.2 Advantages.

1. The working fluid is air from the surroundings so no chemical refrigerants are needed.

2. The open system eliminates any possibility of leakage of the working fluid over time.

B.3 Disadvantages.

1. With two mechanical devices (compressor and turbine) maintenance problems can be substantial. Also, the system reliability decreases because there are more complicated components which may break down. A supply of spare parts would be needed to handle this potential problem.

2. Lubrication seals in the compressor and turbine will degrade with long term storage.

3. The incoming air stream must be filtered to prevent foreign material such as dust, moisture, and oil, from entering the compressor and turbine.

4. Refrigeration COPs are at best modest, and are actually lower than given here, as the computed values given do not account for engine losses in driving the compressor and turbine.

B.4 Tradeoffs, Hazards, and Risks Analysis.

Because air is the working fluid there is no risk from toxic or environmentally unfriendly refrigerants. The system operates at moderate pressures compared to many vapor compression systems, reducing the risks to personnel. Efficiencies are generally lower than found in comparable technologies. Brayton cycle devices are generally used in aircraft air conditioning or similar applications where a supply of pressurized air is readily available for use with no efficiency penalty.

B.5 References.

Holman, J. P., *Thermodynamics*, McGraw-Hill, New York, 1988.

Rolle, K. C., *Thermodynamics and Heat Power,* MacMillan, New York, 1994.

Sisto, F., "The reversed Brayton cycle heat pump - a natural open cycle for HVAC applications", ASME Publication 78-GT-60, 1978.

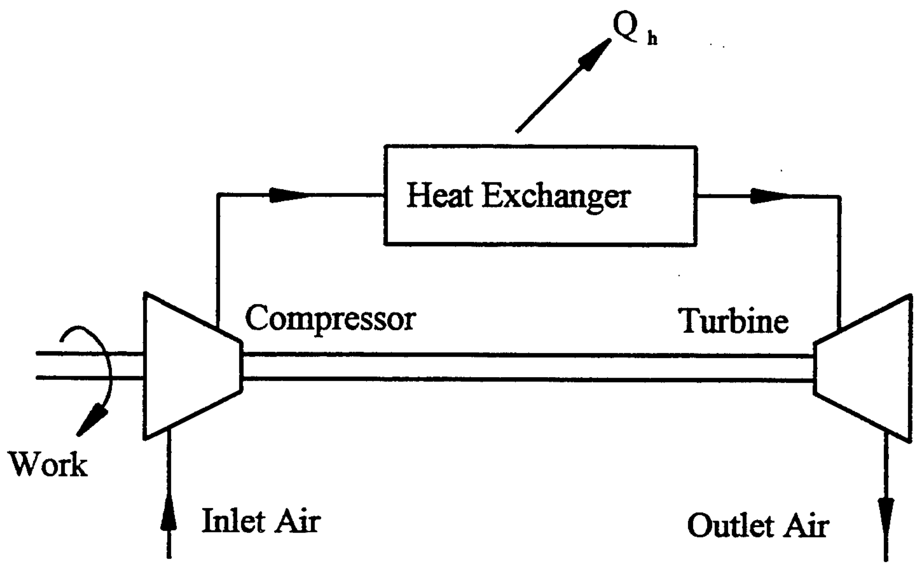

Figure B.1. Open Brayton cycle heat pump schematic.

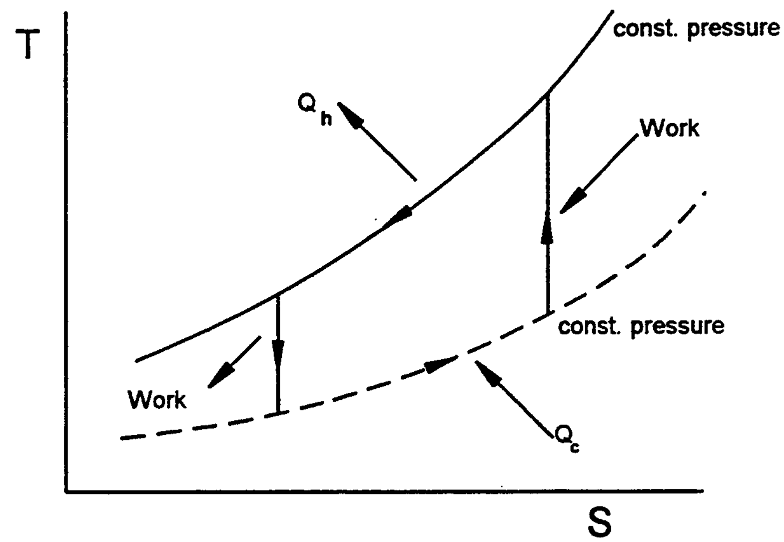

Figure B.2. Open Brayton cycle temperature-entropy diagram.

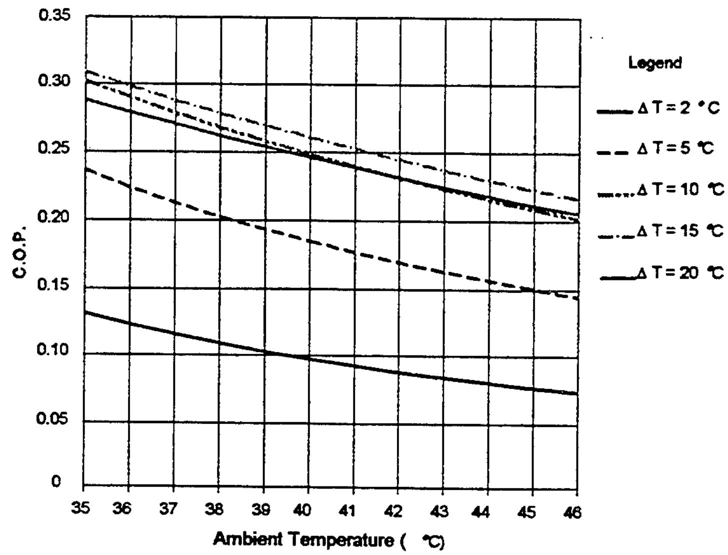

Figure B.3. Open Brayton cycle COP as a function of ambient temperature. Room temperature = 22°C, compressor and turbine efficiencies = 80%.

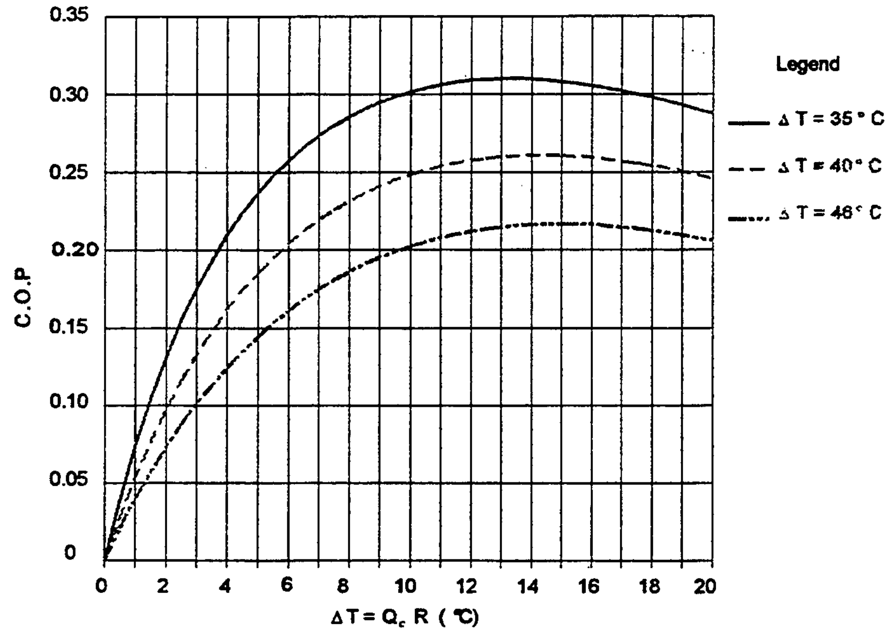

Figure B.4. Open Brayton cycle COP . Curves of constant ambient temperature as a function of ΔT. Room temperature = 22°C, compressor and turbine efficiencies = 80%.

C. Liquid-Vapor Absorption Heat Pumps.

C.1 General Description.

Liquid-vapor heat pumps are partially or wholly heat activated. They are unlike other heat activated heat pumps in that they do not utilize either a compressor or a heat engine as a prime mover. Instead, two fluids are used. One fluid (the refrigerant) is sequentially absorbed, boiled out, condensed, and reabsorbed in the second fluid (the absorbent) to produce the pumping action provided by a compressor in a vapor compression system. Liquid-vapor heat pumps commonly use either lithium bromide and water, or ammonia and water. In the first case, lithium bromide is the sorber and water is the refrigerant. For the second, water is the sorber and ammonia is the refrigerant. A lithium bromide - water heat pump is limited in application by the freezing point of water, and would not meet the storage or operational requirements of the Air Force ECU in cold climates. The ammonia-water heat pump is usable in a much broader range of climates, and will be the only case considered here.

Several versions of the ammonia-water heat pump exist.

C.1.1 Continuous Cycle Absorption Systems.

One of the simplest of heat pumps, with no moving mechanical parts, is a continuous cycle system using a mixture of water, ammonia, and hydrogen under a pressure sufficient to allow the ammonia to condense at room temperature. These systems are sometimes referred to as using the Servel, or Electrolux, cycles. These units consist of boiler, evaporator, condenser, and absorber, all connected in a continuous fashion with no valving separating the various vessels. (See Figure C.1.) The boiler vessel contains a strong solution of ammonia and water. Heat applied to the boiler vaporizes some of the ammonia. The resultant ammonia bubbles carry some of the weak ammonia-water solution upward through tubing which acts as a bubble pump or siphon. The ammonia-water solution is separated from the ammonia vapor by gravity in a section of tubing called the water separator or rectifier, and is routed to the absorber. The ammonia vapor flows to the condenser, where heat is removed and the ammonia vapor becomes liquid ammonia. The liquid ammonia then flows into the evaporator where it is mixed with hydrogen gas to lower the vapor pressure. Heat is added to the mixture from the cold space, and the mixture passes to the absorber vessel. There it is mixed with the weak ammonia solution, with the result that the hydrogen is removed to return to the evaporator. The strong ammonia solution left in the absorber then returns to the boiler, completing the cycle. The flow is operated solely by gravity and pressure differences induced by temperature differences. A thermostat may be used to regulate heat to the boiler, and fans may be used on the condenser. Otherwise there are no moving parts.

These systems are widely used in domestic refrigerators, portable refrigerators, recreational vehicle refrigerators, and in year-round air conditioning of homes and larger buildings. They are heated by electricity, natural gas, propane, kerosene, and a variety of other fuels. They are usually very simple to service and operate, requiring mainly cleaning of the burner and leveling of the system.

C.1.2 Pumped Continuous Absorption Systems.

The previous system can also be operated at higher pressures if a pump is used to maintain the cycle in place of the hydrogen gas. (See Figure C.2.) Again, a strong mixture of ammonia and water in the generator (boiler) is heated, with ammonia vapor passing to the condenser. Heat is removed from the condenser, usually by air cooling. The condensed ammonia passes through a check valve to the evaporator, where it picks up heat from the cold space and evaporates. The ammonia vapor passes through a second check valve to an absorber vessel where it is absorbed to become a strong water and ammonia mixture. A pump then returns this solution to the generator.

The pump and the check valves maintain all of the system except the evaporator at a high pressure, usually in the 200-300 psi range. The evaporator pressure is in the low 40-60 psi range.

C.1.3 Intermittent Cycle Absorption Systems.

The intermittent cycle system consists of a boiler (generator), condenser, liquid receiver, and evaporator vessels (Figure C.3). The mixture used is just ammonia and water. For a prescribed time period heat is added to the boiler which contains a liquid mixture of ammonia absorbed in water. The ammonia is vaporized, and passes upward through the condenser. Here heat is removed, turning the ammonia vapor back to the liquid state. This liquid flows by gravity into the liquid receiver and then into the evaporator. A restriction in the connecting tubing between the liquid receiver and the evaporator insures that the liquid ammonia enters the evaporator slowly. The boiler is then turned off. As the liquid in the evaporator removes heat from the cold space and returns to room temperature in the absence of boiler heating, the ammonia pressure drops and the ammonia is vaporized, allowing it to return by gravity to the generator where it is reabsorbed.

In operation, intermittent heating is the driving force for the cycle. The heating time determines the generating portion of the cycle. At the end of the heating, most of the ammonia is in the liquid receiver and the refrigeration cycle begins. Timing of the heating cycle is determined by the size of the various vessels and the design heating/cooling load.

The pressures generated in this system are appreciably higher than in the continuous system, so welded steel pipes must be used. Again there are no moving parts, except possibly for fans providing cooling air to the condenser. This system can be used in areas where electricity is unavailable, and kerosene or other available fuels can be used as the heat source.

C.2 Analysis of Ammonia-Water Cycle.

The key equations in the analysis of ammonia-water absorption cycles are as follows:

$$\dot{m}_E = Q_E/(h_v - h_f), \qquad \text{(C.1)}$$

where

$\dot{m}_E$ = mass flow of refrigerant from evaporator,
Q_E = cooling load at evaporator,
h_v = enthalpy of refrigerant vapor from evaporator,
h_f = enthalpy of refrigerant liquid from condenser,

and

$$X \cdot WFS_A = (X-1) \cdot WFS_G \text{ for lithium bromide/water,} \qquad \text{(C.2a)}$$

$$X \cdot WFS_A = 1 + (X-1) \cdot WFS_G \text{ for ammonia/water,} \qquad \text{(C.2b)}$$

where

WFS_A = mass fraction of lithium bromide or ammonia in solution from absorber,
WFS_G = mass fraction of lithium bromide/ammonia in solution from generator,
X = mass of solution from absorber per unit mass of refrigerant flow,
X-1 = mass of solution from generator per unit mass of refrigerant flow.

As an example of typical calculations, at an evaporator temperature of 5°C (41.1°F) the enthalpy h_v = 623.02 Btu/lbm for ammonia and at a condenser temperature of 37.8°C (100°F) the enthalpy is h_f = 155.2 Btu/lbm. Thus to have 50,000 Btu/hr of cooling a refrigerant flow rate of 106.8 lbm/hr = 1.78 lbm/min is necessary. For WFS_A = 0.49 and WFS_G = 0.43 (termed a 6% split) the mass of solution from the absorber per unit mass of refrigerant flow is X = 9.5, or after multiplying by the mass rate of refrigerant flow, 16.9 lb/min.

C.3 Advantages.

a. These heat pumps are inherently simple and potentially highly reliable equipment.[1] Solution pumps, blowers, and check valves are typically the only moving parts.

b. The technology is well-known and well-developed.

c. Liquid-vapor sorption heat pumps have lower parasitic energy requirements than do

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[1] In the above discussion and figures, various water traps, heat exchangers, and other auxiliary devices used to improve efficiency have been omitted. These devices are however necessary in many applications to raise efficiencies to acceptable levels, and do contribute to original cost, maintenance, size, and reliability of these systems.

thermal engine heat pumps.

d. Liquid-vapor sorption heat pumps have an inherent energy-storage capability in the form of high-pressure refrigerant vapor.

e. Lithium bromide-water and ammonia-water are the typical fluid-pairs used and are well studied. In lithium bromide-water heat pumps water is the refrigerant, so these heat pumps cannot be used below freezing point of water and are impractical in most heat pump applications, although they are used in nuclear submarines and in some home heating/cooling systems . In ammonia-water heat pumps, ammonia is the refrigerant, so these heat pumps can be used below the freezing point of water.

C.4 Disadvantages.

a. Ammonia is toxic at moderate concentrations (50 ppm) and irritating at low concentrations (3-5 ppm). If the heat pump envelope was breached, much of the ammonia would be released. Being lighter than air, if the heat pump is outdoors and there is a slight wind the ammonia will rapidly disperse. Ammonia is flammable within a small concentration range.

b. Comparatively large heat exchanger sizes are required for an acceptable first cost.

c. The cooling-mode COP is usually lower than that obtainable with Rankine cycle heat pumps. A value for cooling COP around 0.6 is quoted by some manufacturers.

d. The toxicity of ammonia may necessitate the use of a separate heat transfer loop to avoid the possibility of ammonia entering the conditioned place.

e. The corrosivity of ammonia requires the use of steel or aluminum rather than copper, increasing fabrication costs. Synthetic seals must be selected so as to be compatible with ammonia. Joins in steel tubing usually are welded.

f. The need for a generator vessel requires additional equipment complexity, cost, and bulk. The diameter limitation of the generator pressure vessel may require the use of a tall unit for adequate vapor capacity, increasing the bulk of the unit.

g. Sensible heat losses, heats of solution, and the vaporization characteristics of the refrigerant all induce inefficiencies which give COPs below one. Double-effect generators, double-effect evaporation, and internal heat transfer loops can be used to compensate for some of these inefficiencies at the expense of size, complexity, and cost.

h. These systems are heat operated, requiring conversion of electricity to heat for the Air Force ECU.

i. The absorption refrigeration industry is geared to produce much larger size units than are required by the Air Force. Typically cooling in the 50,000 ton range and up is the norm. The present market for condensers and evaporators in the 3-10 ton range is practically nonexistent, although there are no inherent technological reasons holding their development back if a market develops.

j. The initial experience with water-ammonia absorption heat pumps in residences has been called "dismal" due to first cost and equipment reliability factors.

k. Various water traps, heat exchangers, and other auxiliary devices used to improve efficiency are necessary in many applications to raise efficiencies to acceptable levels, and do contribute to original cost, maintenance, size, and reliability of these systems.

C.5 Tradeoffs, Hazards, and Risks Analysis.

One of the chief disadvantages of these devices is their size. Since gravity is an important factor in their operation a minimum height of 40 inches or more is necessary. To improve efficiency, add-on devices could be used which would increase the height further.

It is not clear how well these devices are suited to long-term storage in harsh environments. In cold temperatures the water could freeze, and in hot temperatures the pressures will be high, requiring extra weight in the vessels and connecting pipes. Also, it may be necessary to store them vertically, or to provide for a means of starting the heat pump when they have been upended.

Because of the importance of gravity to their operation these heat pumps must be installed on a solid base so that they are close to vertical. This will increase setup time and the base pad will be an extra weight.

Since ammonia is toxic, external heat transfer loops on the condenser and evaporator are needed. To keep the heat exchanger sizes small, these loops will have to use a liquid as the heat transfer fluid, requiring pumps and extra setup time.

C.6 Manufacturers.

Bernzomatic, Inc., 1 Bernzomatic Drive, Medina NY 14103, 716-798-4949.

Dometic Inc., 2320-T Industrial Parkway, PO Box 490, Elkhart IN 46515, 219-294-2511.

Electrolux AB, 5-105 45, Stockholm Sweden, XY6-8-7386000.

C.7 References.

Althouse, A. D., C. H. Turnquist, and A. F. Bracciano, *Modern Refrigeration and Air Conditioning*. Goodhart-Willcox, South Holland IL, 1992.

American Society of Heating, Refrigeration, and Air-Conditioning Engineers, *Fundamentals*, ASHRAE, Atlanta GA, 1993.

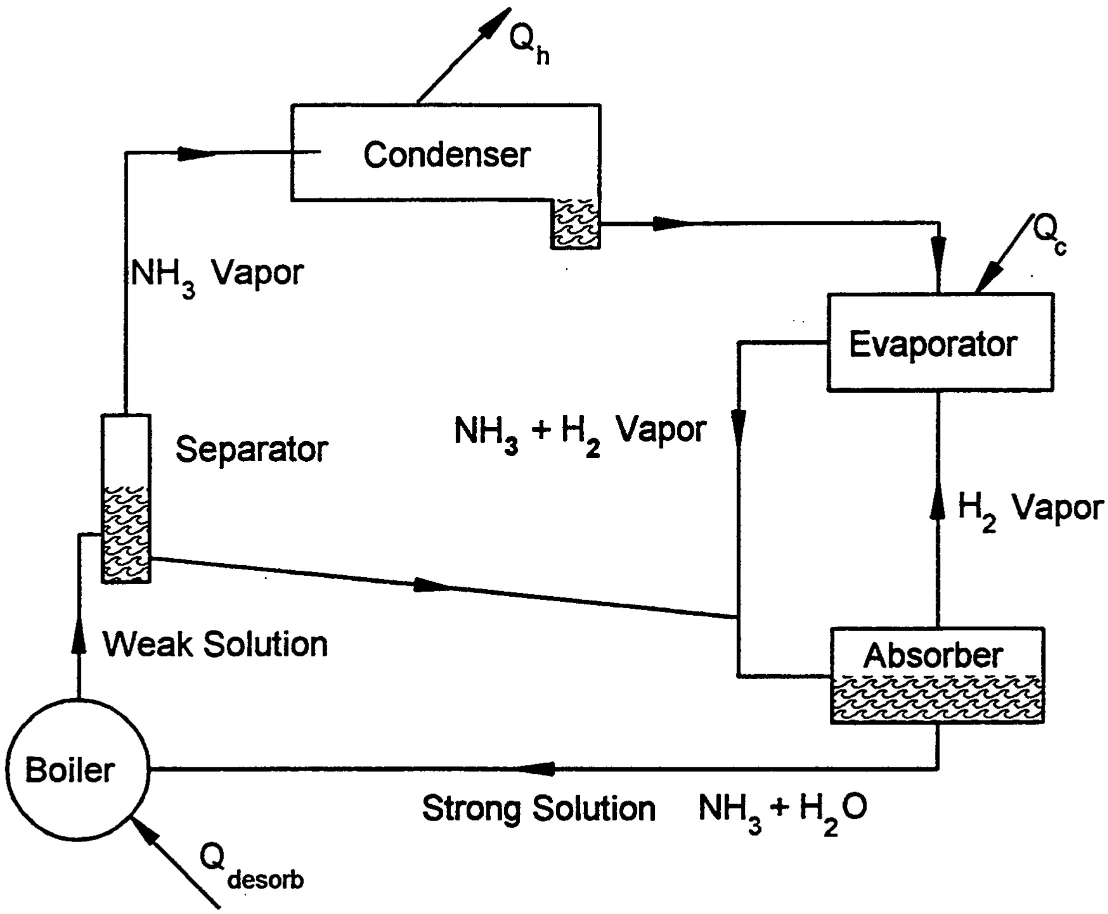

Figure C.1. Continuous cycle liquid-vapor absorption heat pump.

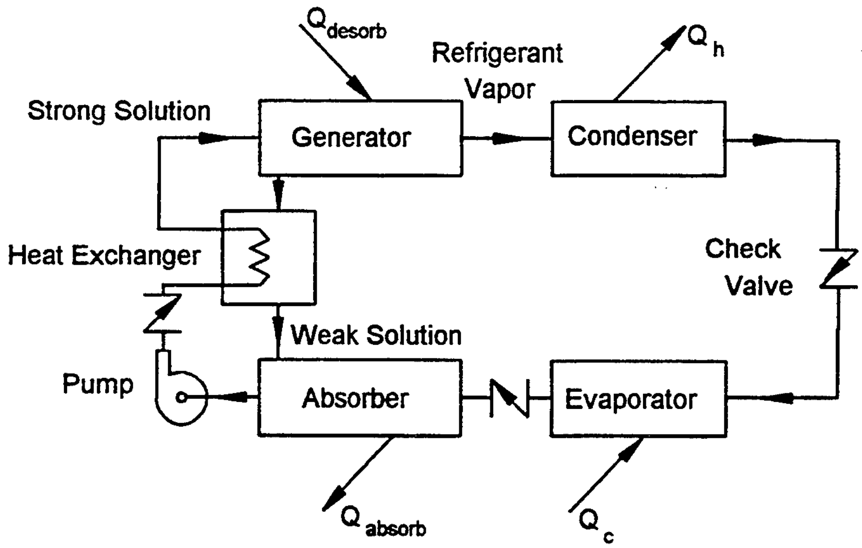

Figure C.2. Pumped continuous cycle liquid-vapor absorption heat pump.

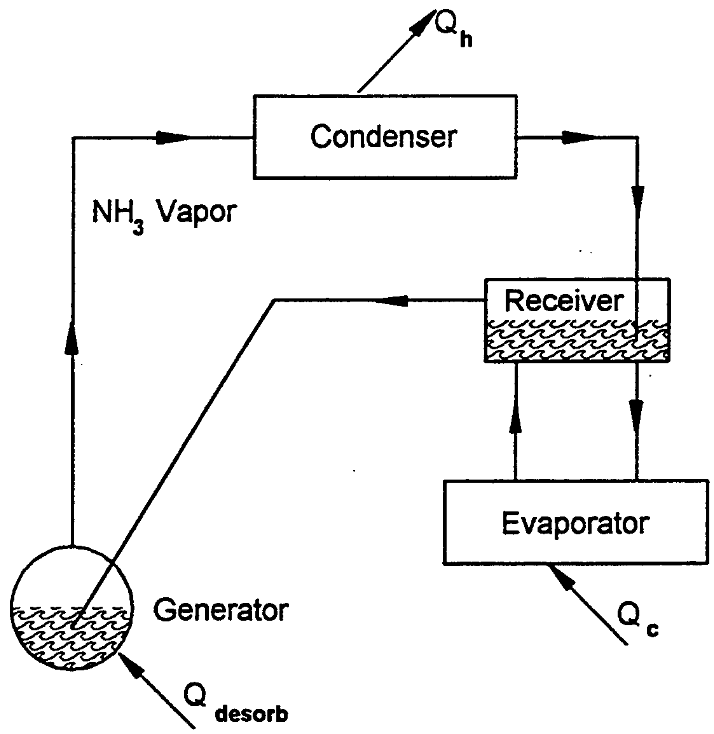

Figure C.3. Intermittant cycle liquid-vapor absorption heat pump.

D. Magnetic Heat Pumps.

D.1 Description.

A magnetic heat pump operates on the principle that many magnetic materials become warmer when in the presence of a magnetic field, and cooler when the magnetic field is removed. In effect, the magnetic material plays the role of the working fluid or refrigerant in a heat pump.

To date, most magnetic heat pumps have been used at extremely low temperatures, typically around 10 K or less. Recently, the rare earth element gadolinium has been used for heat pumps at near room temperatures. Gadolinium can change in temperature as much as 14°C when a strong magnetic field is applied. Using regeneration, temperature changes in gadolinium higher than 60°C have been achieved.

An early model of a magnetic heat pump was built at NASA's Lewis Center in the late 1970's. The gadolinium refrigerator section was made up of thirty-six 1 millimeter thick plates contained within a stainless steel cylinder. This cylinder was in turn suspended in a fixed position within the regenerator. The regenerator was a 0.1 meter long, 50 millimeter diameter fluid-filled cylinder which was caused to move up and down by a motor. (See Figure D.1.) The fluid used inside the regenerator was a mixture of 50% water and 50% alcohol. The regenerator was placed within the field of an electromagnet in such a manner that the gadolinium was at the center of the magnetic field. The magnet used was a superconducting magnet.

Other configurations of magnetic heat pumps have accomplished the above cycle by using staged valving together with a number of regenerators, so that the fluid is directed to a particular regenerator depending on the state of the magnet on/off cycle. Still other configurations have been devised using either linear reciprocating or rotary motion, and static magnetic material with charging and discharging magnets. A cycle without regeneration was accomplished by rotating a disk containing a number of gadolinium segments insulated from one another between a stationary heat exchanger and a ring which has a magnetic sector. The latter configuration avoids the need for flexible heat transfer connectors which would be subject to thermal cycling and thus mechanical fatigue. It has been stated in the literature that nonregenerative magnetic heat pumps are limited to a temperature change of 15°C at temperatures below 20 K, and an even smaller temperature range at higher temperatures.

At very low temperatures (less than 20 K), the magnetic heat pump cycle closely approaches the Carnot cycle. Above 20 K the cycle rapidly loses efficiency. In the very low temperature range paramagnetic (antiferromagnetic) magnets are used. At higher temperatures, magnetic heat pump cycle efficiency lies somewhere between the Ericsson and Brayton cycles. At these temperatures, ferromagnetic or ferrimagnetic magnets near their Curie point temperature must be used. (The Curie point temperature is the temperature above which a substance possesses a spontaneous magnetic moment, even in the absence of an applied magnetic field.) Typical Curie point temperatures are 1,131°C for cobalt, 770°C for iron, 358°C for nickel, and 16°C for gadolinium.

At this time, it is not possible to evaluate the efficiency of the various magnetic heat pumps which have been constructed, as detailed data on all of the energy sources needed to operate these devices is not available in the open literature. It may be that superconducting magnets are necessary for the device's successful operation as an ECU. If so, this would add considerably to the overall size, weight, and complexity, with an attendant loss in efficiency. It appears certain that until much better magnetic materials are available, both for the refrigerant and the field-producing magnet, a realistic efficiency would be extremely low for an overall system.

D.2 Tradeoffs, Hazards, and Risks Analysis.

The necessity of having superconducting magnets adds a considerable amount of support equipment to the heat pump. So far these heat pumps apparently have not been used outside of the laboratory, where their application is in the cryogenic temperature range. This presently is not a serious candidate for the Air Force ECU.

D.3 References.

Angrist, S. W., *Direct Energy Conversion*, 3rd ed., Allyn & Bacon, Boston, 1976.

Barclay, J.A., "Magnetic refrigeration: a review of a developing technology", Adv. in Cryogenic Engring. Proc. 1987 Crogenic Engring. Conf., vol. **33**, pps. 719-731, 1987.

Barclay, J.A., & W.A. Steyert, EPRI Final Report EL-1757, April 1981.

Brown, G.V., "Magnetic heat pumping near room temperature", J. Appl. Phys. vol. **47**, pps. 3673-3680, 1976.

Brown, G.V., ASHRAE Trans. vol. **87**, p. 783 , 1981.

Hakuraku, Y., "Thermodynamic simulation of a rotating Ericsson-cycle magnetic refrigerator without a regenerator", J. Appl. Phys. vol. **62**, pps. 1560-1563, 1987.

Jaeger, S.R., J.A. Barclay, & W.C. Overton jr., "Analysis of magnetic refrigeration with external regeneration", Adv. in Cryogenic Engring. Proc. 1987 Cryogenic Engring. Conf., vol. **33**, pps. 751-755, 1987.

Patton, G., G. Green, J. Stevens, & J. Humphrey, Proc. 4th Int. Cryocooler Conf., Sept. 1986.

Rosenblum, S. S., W. A. Steyert, & W. P. Pratt jr., Los Alamos National Laboratories, Report LA-6581, May 1977.

Sauer, H.J. & R.H. Howell, *Heat Pump Systems*, J. Wiley, New York, 1983.

Wilson, M.N., *Superconducting Magnets*, Clarendon Press, Oxford, 1983.

Wood, M.E., & W.H. Potter, "General analysis of magnetic refrigeration and its optimization using a new concept: maximization of refrigerant capacity", Cryogenics vol. **25**, 1985.

Yan, Z., & J. Chen, "The effect of field-dependent heat capacity on the characteristics of the ferromagnetic Ericsson refrigeration cycle", J. Appl. Phys. vol. **72**, pps. 1-5, 1992.

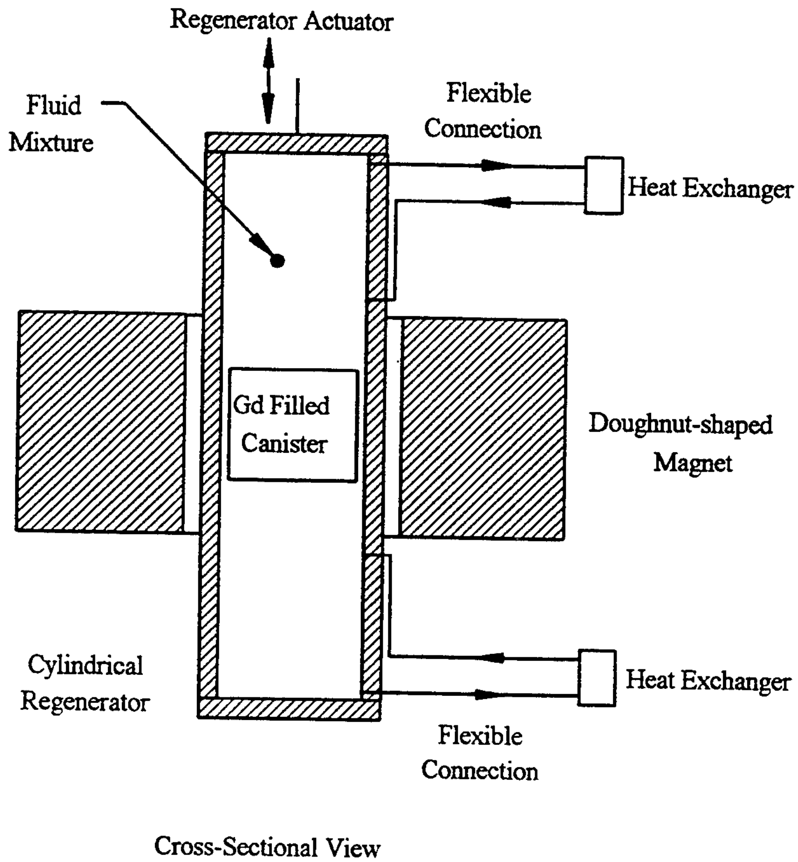

Figure D.1. Magnetic heat pump schematic (as built by NASA-Lewis Center).

E. Malone-Stirling Heat Pumps.

E.1 Discussion.

Engines and heat pumps using liquids rather than gases in Stirling cycles were first proposed by Malone in the 1920's and early 1930's [Malone 1931]. Their study has recently been taken up by Swift at Los Alamos National Laboratories [Swift 1989a and b], who performed a theoretical analysis and built an experimental engine. Malone engines operate very much as do gas Stirling engines, with the same heat exchangers and regenerators. Thus the inefficiencies and problems which have hindered the introduction of Stirling engines would seem to apply to these engines as well. At this time the published literature shows little followup of these studies, so there has been little critical evaluation or discussion of the concept.

Except for engines built by Malone in the 1920's and 30's, and an experimental engine by Swift, no other development of this concept has been found under this study. Malone's drawings of his engines are reminiscent of nineteenth century drawings and are difficult to follow. An adaptation of Swift's design is shown in Figure E.1. The crankshaft shows two fixed phase shifts of 90° and a variable phase shift ϕ, where $0° \leq \phi \leq 40°$. Four bypass valves were included in the demonstration heat pump for control and measurement purposes. Heat fluxes of 1.3 kilowatts at pressures of 57 bars (838 psia) and a drive frequency of 5 hertz (300 rpm) were obtained.

E.2 Advantages.

a) Liquids have a better heat transfer efficiency than gases. Their heat capacities per unit volume (ρc_p) are orders of magnitude larger than those for gases. This means that their heat exchangers can be very compact and require smaller pumping power than for a heat pump which uses gas as a working fluid.

b) Use of a liquid rather than a gas greatly reduces explosion risk because stored internal energy is low.

c) Liquids have larger thermal expansion coefficients than gases over the temperature range 0.7 $T_{critical}$ to $T_{critical}$. Thus if a liquid can be found whose critical temperature is near the desired operating temperatures of the application, the thermal expansion performance of the liquid can be better than that of a gas under the same temperature conditions.

d) Liquid thermal expansion is greater than liquid compressibility. This allows great changes of pressure with small relative changes in temperature. In particular, temperature changes accompanying adiabatic pressure changes are very small.

e) Since temperature changes are small, the Malone engine can in principle operate at much lower temperatures than a Rankine engine operating at the same output power conditions. Thus all joints and glands will be cold, eliminating hazards, the need for cumbersome insulation, and lowering potential leakage problems.

f) Malone heat pumps have the potential to provide a given heating/cooling load from a smaller volume than a comparable Rankine engine because of the need for smaller heat exchangers.

g) Flexibility of operation and design has been claimed, but this has not been discussed or elaborated on in the open literature.

E.3 Disadvantages.

a) Theoretical efficiencies of 0.7 Carnot efficiency have been claimed at operating temperatures of $T_{cold} = -15°C$, $T_{hot} = 30°C$, and an oscillatory pressure amplitude of 30 bars. Such a large oscillatory pressure amplitude would mean a maximum pressure greater than 60 bars (880 psi). Models built by Swift however show maximum efficiencies of only 0.2 Carnot efficiency, with oscillatory pressure amplitudes of 41 bars. This high an oscillatory pressure means that the maximum pressure will be greater than 82 bars (1,205 psi). Such large pressures require substantial cylinder walls and tubing, and any liquid leaking at such pressures would be a potential hazard to operators.

E.4 Tradeoffs, Hazards, and Risks Analysis.

The high pressure levels required for an ECU, higher than is standard in Rankine cycle engines, introduce personnel hazards and risks which would appear to be substantially outweigh any of the advantages claimed. While water was used as the refrigerant by Malone, Swift apparently used propylene in his model which is not as environmentally friendly or hazard free. Propylene has a critical temperature of 90°C (194°F) and a critical pressure of 46 bars (88 psia), which makes it suitable for ECU applications. Swift found that bearing and seal drag, open-surface pressurization, and piston leakage accounted for 66% of his estimated inefficiencies. Open-surface pressurization is his term for losses associated with the oscillatory heat flow into and out of the solid stationary portions of the heat pumps. While the losses reported by Swift are for an exploratory engine and could be reduced, they are losses of the types which have plagued the development of the Stirling engine, and their reduction has not proved to be a trivial task.

There have been few further studies of this engine reported, and it is reasonable to assume that at this time the development of the Malone engine is substantially behind that of the Stirling engine. Since the form an actual embodiment of such an engine might take would be largely speculative at this point, no further analysis of the Malone cycle has been made in this study.

E.5 References.

Malone, J. F. J., "A new prime mover", J. Royal Soc. Arts, London, vol. **79**, pps. 679-709, 1931.

Swift, G. W., "Simple theory of a Malone engine", Proc. 24th Intersoc. Energy Conversion Engring. Conf., pps. 2355-2361, 1989.

Swift, G. W., "Experiments with a Malone engine", Proc. 24th Intersoc. Energy Conversion Engring. Conf., pps. 2385-2393, 1989.

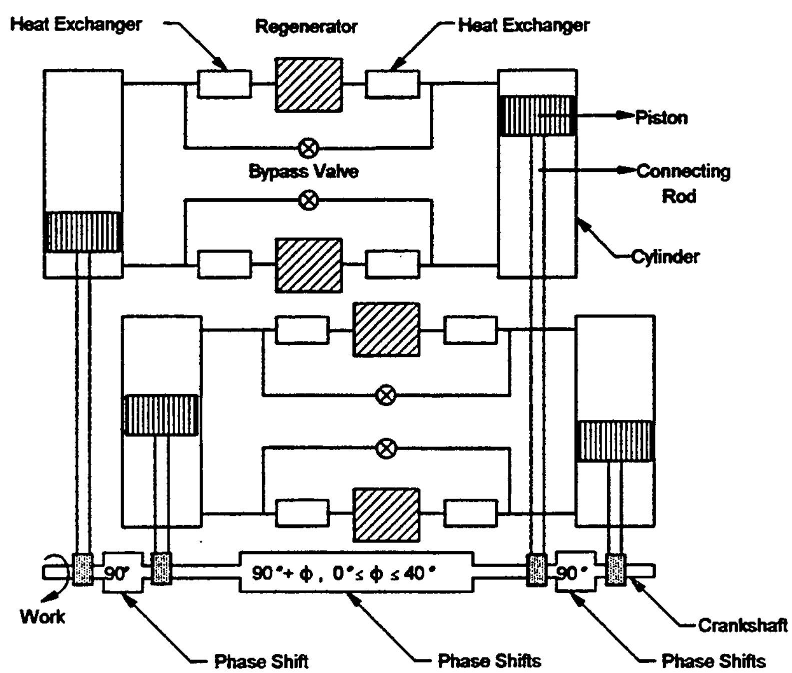

Figure E.1. Four-cylinder Malone-Stirling heat pump schematic.

F. Solid-Vapor Absorption Heat Pumps.

F.1 General Description.

Solid-vapor absorption heat pumps are similar to vapor-compression absorption heat pumps in many respects, and can be thought of as a standard vapor-compression refrigeration cycle with the mechanical compressor replaced by a chemical compressor. The embodiment of solid-vapor absorption heat pumps includes a vessel containing the sorbent compound being desorbed, condenser, throttling process, evaporator, and a second vessel containing the same type of sorbent as the first vessel but used for absorbing [Herold and Rademacher 1995]. The cycle is not continuous, but rather is accomplished by interchanging the two sorbent vessels by means of valving after they are desorbed/absorbed. Activated carbon, metal hydrides, and zeolites have all been used as sorbents, but so far have shown relatively low values for maximum useable refrigerant stored per unit mass of sorbent. This is because the refrigerant tends to be first absorbed on the surface of these absorbers, building up a surface layer which hinders absorption in the interior of the absorber [Rockenfeller 1985, 1987]. Thus these absorbers must be manufactured in the form of thin layers of the compound on a heat exchanger surface. Metal hydrides also have had the problem of forming a grey powder which blocks and contaminates the heat pump passages.

Recent techniques developed for complex compounds [Graebel et al 1991] provide possibilities of refrigerant concentrations an order of magnitude higher compared to previous methods. This has been achieved by preprocessing the complex compound using a proprietary technique which allows the refrigerant to diffuse rapidly throughout the sorber with no buildup of a surface layer. Mass transfer of the refrigerant gas and heat transfer to the sorbent are thereby enhanced. For this reason complex compound heat pumps are the more developed and most promising of the absorption technologies at this time.

F.2 Complex Compound Chemistry.

A complex compound consists of a compound formed by an inorganic salt (often the chloride or flouride salt of a metal) bonded with a non-metallic gaseous molecule (often ammonia or water), the gaseous molecule being called the ligand [Parker 1974]. In the bonding process, these complexes orient themselves in a polar structure to form a coordinative covalent bond between the ligand and the absorbent salt. Since the coordinative covalent bond is formed from the sharing of electrons, the bond can be formed and broken reversibly. Several ligand molecules can be bonded to a single salt molecule, so that the amount of ligand mass which can be bonded is typically four or five times as high as could be expected from either a bond-breaking reaction or physical absorption. Ammines[1] are the most promising candidates as salts for refrigeration systems using complex compounds.

[1]Ammines are coordination compounds formed by the union of ammonia with a metallic salt such as sodium chloride or calcium chloride in such a way that the nitrogen atoms of the ammonia are directly linked to the metal ion.

The bonding process between the ligand and the salt occurs in a number of coordination steps. Within a given coordination step, the vapor pressure and heat of reaction of the compound are virtually independent of the ligand concentration within that coordination sphere (i.e., the valence shell of the salt). Thus, a partially filled coordination sphere has the same vapor pressure as a completely filled coordination sphere. The complex compound can be made to adsorb or desorb completely within the coordination step at a constant temperature simply by adjusting the pressure of the refrigerant surrounding the salt to be above or below the complex compound's vapor pressure. This is in contrast to usual physical adsorption processes, where a change in the sorbate concentration results in a change in its vapor pressure. Besides the advantage of the simplicity of a sorption process occurring at a constant pressure, this means that there is a tendency for the salt to saturate to its maximum capacity as long as the pressure is held constant. In contrast, during physical adsorption there is a tendency to reach an equilibrium concentration on the surface of the sorbent, which need not be the sorbent's maximum capacity. Consequently, under most conditions a complex compound can be expected to have a much higher mass-sorbing capacity than a sorbent undergoing physical sorption.

To clarify the nature of these coordination steps, consider an experiment run at constant temperature where the unabsorbed complex compound is exposed to the ligand. As the ligand pressure is slowly raised, a pressure level would be reached where an ideal complex compound would suddenly start to absorb the ligand, the absorption continuing at constant pressure until N_1 moles of the ligand have been absorbed per mole of salt. N_1 is termed the "width" of the first coordination step. Once the step has filled, absorption ceases. If the ligand pressure is slowly increased further, a new pressure level would be reached where the complex compound would absorb N_2 more moles of ligand per mole of salt, N_2 being the width of the second step. This process would repeat further, eventually being limited by the saturation curve of the ligand. The ideal complex compound would have abrupt steps as described here - a real complex compound would have the steps rounded so as to be less abrupt.

The dissociation energy (i.e., the energy required to cause formation or breakage of the bonds between salt and refrigerant) of a complex compound depends largely on the ligand. For heat pump and refrigeration applications, to minimize the physical size of the heat pump and to enhance heat and mass transfer it is important to achieve a high ratio of mass storage of the ligand per unit mass of salt. Ligands which have only one atom available for bonding usually have lower bond energies than those with more available atoms. This leads to ammonia and water as the most promising ligands. Other important major selection criteria for ligands are price, toxicity, chemical stability, corrosivity, and having vapor pressures in the relevant temperature range of the application.

For solid-vapor metal inorganic complex compounds such as hydrates and ammines ligand vapor pressure is a function of the temperature of the complex compound. As stated above, within one coordination step the vapor pressure is independent of the molar ratio of adsorbent to adsorbate, and a particular mole of salt can absorb up to, say, N moles of ammonia, N being the step width. A generic phase diagram for both ammonia and a complex compound (CC) within this step is shown in Figure F.1. The reaction for a given step is given by

$$CC \cdot N(NH_3) \rightleftharpoons CC + N\,(NH_3). \qquad (F.1)$$

Within a step the equilibrium pressure of a complex compound is related to the temperature by a formula first given by Nernst [Nernst 1969, Wilkinson 1980, Bougard et al 1994]. A simplified form of the Nernst formula is

$$\log p_{ligand} = \frac{-Q_o}{2.303\ RT} + a. \qquad (F.2)$$

Here Q_o represents the heat needed to break or form the coordination bond, R is the universal gas constant, and "a" is a constant of the coordination step. Both Q_o and "a" are determined experimentally for a given ligand [Kober 1979].

The saturation curve for a higher coordination step lies above the lines for lower coordination steps for the same complex compound. When the logarithms of ammonia saturation pressure and complex compound vapor pressure are plotted versus inverse temperature, the plots are nearly straight lines. Such plots are called either van t'Hoeft plots or Clapyron diagrams. Figure F.1 shows equilibrium curves for a typical salt having a number of coordination steps, as well as the saturation curve for ammonia. The reference by Bougard et al [1994] shows a number of equilibrium curves for salts usable with ammonia.

Sorption processes occurring in solid-vapor heat pump and refrigeration cycles are very complicated and are difficult to describe in a mathematical model. Heat and mass transfer properties of the salt and its supporting heat exchangers are more important than surface sorption processes. The physical structure of the salt may change during the initial sorption, affecting both heat and mass transfer. Later sorptions may also result in structural changes of the salt, but less likely so than during the initial sorption. At a given pressure the rate at which the ligand is adsorbed initially grows linearly in time and then decreases until the coordination step is filled. To date it has been impossible to predict the rate at which sorption takes place. Thus the behavior of each solid complex compound must be experimentally investigated separately for each application within the desired temperature and pressure limits.

F.3 Complex Compound Heat Pumps.

The principle of operation of a complex compound heat pump is shown in Figure F.2. Single arrows indicate the direction of refrigerant flow, and double arrows indicate the direction of heat flow. The log pressure and inverse temperature plot illustrate the thermodynamic behavior. The operation of the cycle is as follows:

1-2 The complex compound in vessel A is heated with prime energy at the desorption temperature T_1. This results in a ligand vapor pressure higher than the pressure of the ligand (shown in Figure F.2 as NH_3) at the condenser temperature T_2. The salt in vessel A therefore undergoes an endothermic desorption reaction and the ligand is cooled to the condenser temperature, in the process changing phase from a vapor to a liquid. The condensing reaction is an exothermic process, releasing thermal energy at the heating temperature T_2.

2-3 The condensed ligand is next cooled to the refrigeration temperature T_3, typically by passing through an expansion valve to an evaporator. Heat is added to the evaporator resulting in the ligand changing back to the vapor phase.

3-4 The vapor pressure of the ligand at temperature T_3 is higher than the vapor pressure of the complex compound in vessel B at temperature T_4. Thus the ligand leaves the evaporator and is absorbed onto the salt in vessel B, releasing energy at the rejection temperature T_2 (approximately equal to T_4) as it does so. Complex B is the same as complex A, but is in a separate vessel.

4-1 Complex compounds A and B are "interchanged" by cooling A to temperature T_4 and heating B to temperature T_1. As part of the interchange switching of the refrigeration lines also takes place. The cycle is now ready to repeat.

As is shown in Figure F.2 the complex compound occupies two vessels. This allows nearly continuous operation of the heat pump. During the first half of a cycle, the complex compound in one vessel is being desorbed, and the complex compound in the second vessel is being adsorbed. In the last half of the cycle, valving interchanges the vessels and the roles of the two vessels are reversed. A short time period is necessary between the two halves of the cycle to allow cooling of one vessel and heating of the other.

The coefficient of performance (COP) of the above single-stage heat pump is not high. Theoretically, a refrigeration COP in the range of 0.6 to 0.7 should be possible, as shown in Figures F.3 and F.4. The ΔT shown is a measure of the evaporator heat transfer efficiency, being the temperature drop from the cool environment to the ammonia in the condenser. These COPs are computed according to

$$COP_{refrigeration} = Q_{evaporator}/(Q_{subcooler} + Q_{desorb}). \qquad (F.3)$$

This assumes that all of the heat needed to raise the temperature of the salt, supporting heat exchanger, and containing vessel can be recovered. Since the vessel will generally have a high heat capacity, this is not likely. A more realistic expression for the COP is

$$COP_{refrigeration} = Q_{evaporator}/(Q_{unrecoverable} + Q_{subcooler} + Q_{desorb}) \quad (F.4)$$

where $Q_{unrecoverable}$ is the amount of this heat which cannot be recovered. Including this factor in the COP cuts it by nearly half.

Heating COPs are shown in Figures F.5 and F.6. Since for heating purposes much of the heat used in heating the vessels has the potential of being recoverable, these COPs are more realistic than the cooling COPs.

A three stage cycle has been proposed [Rockenfeller and Kirol 1994] which uses three different salts in three vessels with regenerative heat transfer occurring between the vessels. In the first part of the cycle shown in Figure F.7, ammonia is desorbed from complex compounds 1 and 3 and passed to the condenser. Complex compound 2 absorbs heat from the evaporator. At the high pressure side of the circuit complex compound 1 is heated and the condenser temperature is lowered. At the low pressure side, the evaporator is heated and heat given off from complex 2 absorption is passed to complex compound 3. In the second part of the cycle, complex compound 2 desorbs to the condenser and complex compounds 1 and 3 absorb from the evaporator. At the high pressure portion of the circuit heat is extracted from the condenser. At the low pressure portion, heat is added to the evaporator and rejected from complex compounds 1 and 3. The heat rejected from complex compound 1 is used to desorb complex compound 2. A possible embodiment of such a heat pump is shown in Figure F.8. One way valves are used in the ammonia circuit. The heat transfer circuits would be actively controlled either through a timing system or a control system activated by temperature and pressure sensors.

F.4 Advantages.

a. Complex compounds have high refrigerant holding capacities per mass of salt.

b. All heat absorption and rejection takes place at nearly constant temperature.

c. A large selection of complex compounds is available, making possible almost any desired vapor pressure curve to suit a given application.

d. A high energy per unit of refrigerant adsorbed is possible, exceeding 1 kilowatt per kilogram of adsorbent.

e. Staging techniques allow increased efficiency for a given temperature lift. Staging is only limited by the available firing temperature.

f. Extremely high and extremely low pressures are avoided.

F.5 Disadvantages.

a. Ammonia is slightly irritating at levels of 3 to 5 parts per million (ppm), and quite irritating at 15 ppm. At 30 ppm respirators are needed, and OSHA limits exposure at 50 ppm to 5 minutes. At 50,000 ppm ammonia is hazardous to life. It is flammable within the range of concentrations 150,000 to 270,000 ppm.

b. In the single stage heat pump two chambers of complex compounds are needed, one for adsorbing and one for desorbing. This increases system size, complexity, and weight.

c. The system must be run in a staged rather than a continuous manner, thus lowering the potential heat pumping. Complex compound cooling and heating loops must be switched at the end of a cycle.

d. The system is heat driven. For the Air Force ECU this means that electric power must first be converted to heat. A pumped liquid loop must be provided to alternately heat and cool the complex compound chambers, with attendant heat losses.

e. The long-term stability of complex compounds under severe handling and storage conditions is as yet unproven.

f. The use of a 3 stage cycle to improve efficiency necessitates heat transfer loops between the salt vessels. These loops require either pumps or thermosiphons, adding to the size, complexity, and weight of the heat pump.

g. Both heating and cooling of the reactor vessels is necessary for sorption. While heating can be done electrically, a liquid heat transfer loop is necessary for the cooling.

h. In practice, a control system may be necessary to monitor pressures, temperatures, and flow rates. Such a system would be quite complicated and expensive.

i. Only a few single stage heat pumps have been built. At this time it appears that no three cycle heat pumps have been built.

F.6 Tradeoffs, Hazards, and Risks Analysis.

Both toxicity and flammability are potential problems with these heat pumps because of the use of ammonia as a refrigerant. The ammonia refrigeration industry however has developed reliability of such systems to a high level, with few accidents reported. The use of complex compounds reduces the risk further, as the amount of ammonia in the system is within OSHA safety standards for the home (less than 10 pounds), and approximately half of the ammonia is absorbed on the salt, reducing the available ammonia for leakage. The pungent odor of ammonia alerts personnel to even the slightest leak (1 or 2 parts per million is easily detectible), and only a major leak of ammonia would be a hazard. Since ammonia is lighter than air it disperses rapidly. The range of concentrations in which ammonia is flammable should not be experienced since the heat pump would normally be exterior to the structure.

The high internal pressures pose a risk to personnel, and increase the weight of the system. Pressures are typically in the range of 15 bars (220 psia) and higher.

Temperatures for desorption are also high. Thus the thermal signature is high. Since there are no moving mechanical parts these heat pumps are silent in operation.

The minimum active control system for these heat pumps would be a clock which switches the absorbed and desorbed vessels at the end of the sorbing period. This clock would open and close solenoid valves on the heat transfer loops to the vessels. Heating could be accomplished directly by electrical resistive heating, but cooling would almost certainly require a liquid heat transfer loop. In principle, switching of the ammonia lines can be accomplished passively by the use of one way flow valves, provided that such valves function reliably in the presence of ammonia over a range of temperatures and pressures.

At this time there have been several small prototypes built and tested, although the test results are not complete and are not available in the open literature. The technology shows much promise, but until a number of prototypes have been built, tested, and operated over a reasonably long length of time the future of the technology is not clear.

F.7 Manufacturers.

None at this time. Rocky Research, 1598 Foothill Drive, Boulder City NV 89006, 702-293-0851 has been active in the research and development of this technology and has licensed some aspects of their results to other companies.

F.8 References.

Bougard, J., R. Jadot, & V. Poulain, "Solid-gas reactions applied to thermotransformer design", Proc. Int. Absorption Heat Pump Conference, ASME AES-vol. 31, pps 413-418, 1994.

Graebel, W. P., U. Rockenfeller, & L. Kirol, "Solid-vapor sorption refrigeration systems", 13th Indust. Energy Tech. Conf., pp. 43-48, 1991.

Herold, K, & L. Radermacher, *Absorption Chillers and Heat Pumps*, CRC Press, Boca Raton, 1995.

Kober, F., *Grundlagen der Komplexchemie*, Otto Salle Verlag, 1979.

Nernst, W., *The New Heat Theorem*, Dover Publications Inc., New York, 1969.

Parker, S. P., ed., *Dictionary of Scientific and Technical Terms*, 4th edition, McGraw-Hill Book Co., New York, 1974.

Rockenfeller, U., "Entwicklung eines Niedertemperatur-Speicher-wärmepumpen-systems für Kühlung und Heizung", RWTH Aachen, Germany, 1985.

Rockenfeller, U., "Study of Generic Problems of Solid-Vapor Energy Storage Systems", Oak Ridge National Laboratory, subcontract #86X-57432V, Final Report, 1987.

Rockefeller, U., & L. D. Kirol, " HVAC and heat pump development employing complex compound working media", Proc. Int. Absorption Heat Pump Conference, ASME AES-vol. **31**, pps 433-437, 1994.

Wilkinson, F., *Chemical Kinetics and Reaction Mechanisms*, Van Nostrand-Rheinhold, 1980.

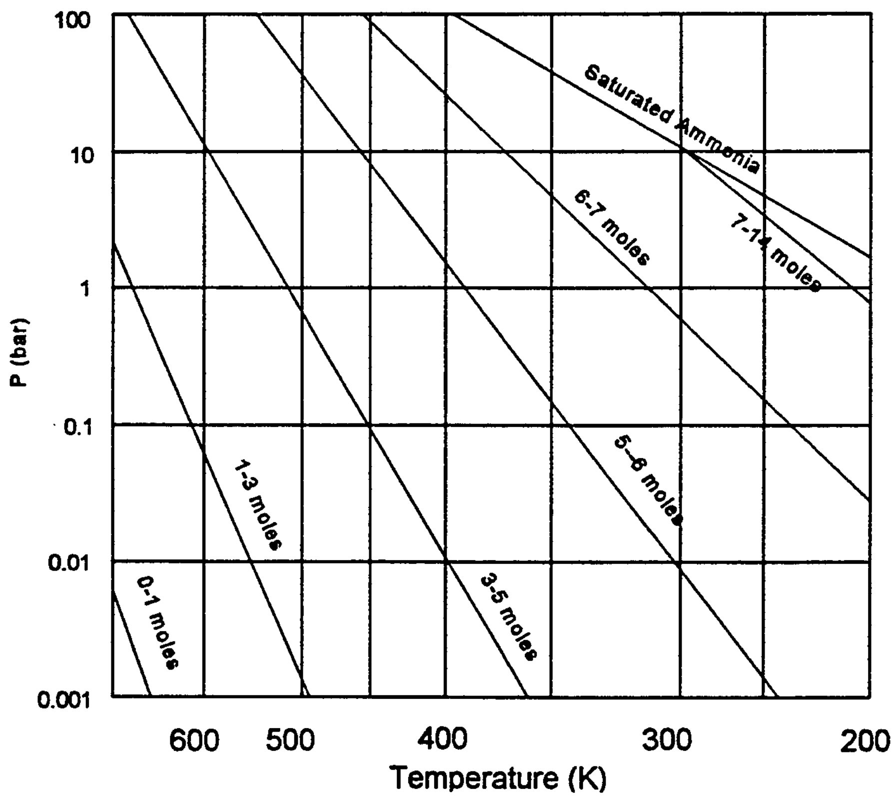

Figure F.1. Saturation curves for a salt with a number of coordination steps. The ammonia saturation curve is included.

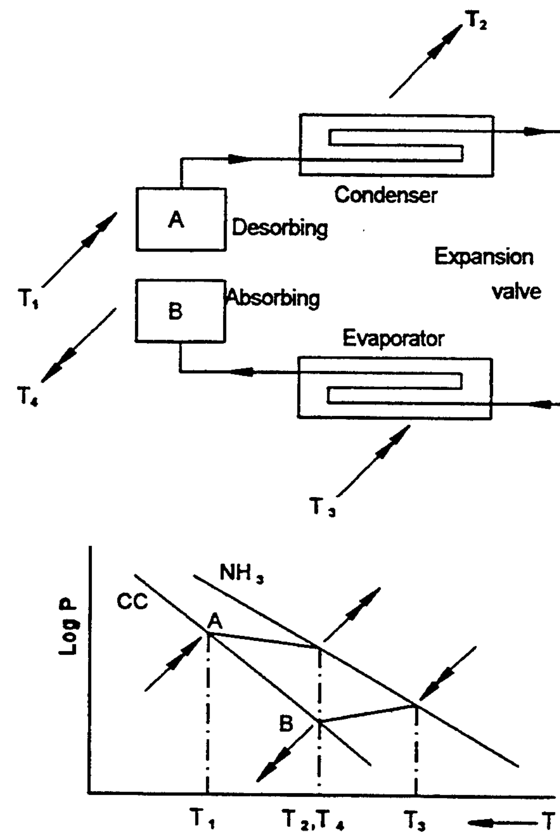

Figure F.2. Single-stage complex compound heat pump cycle and schematic.

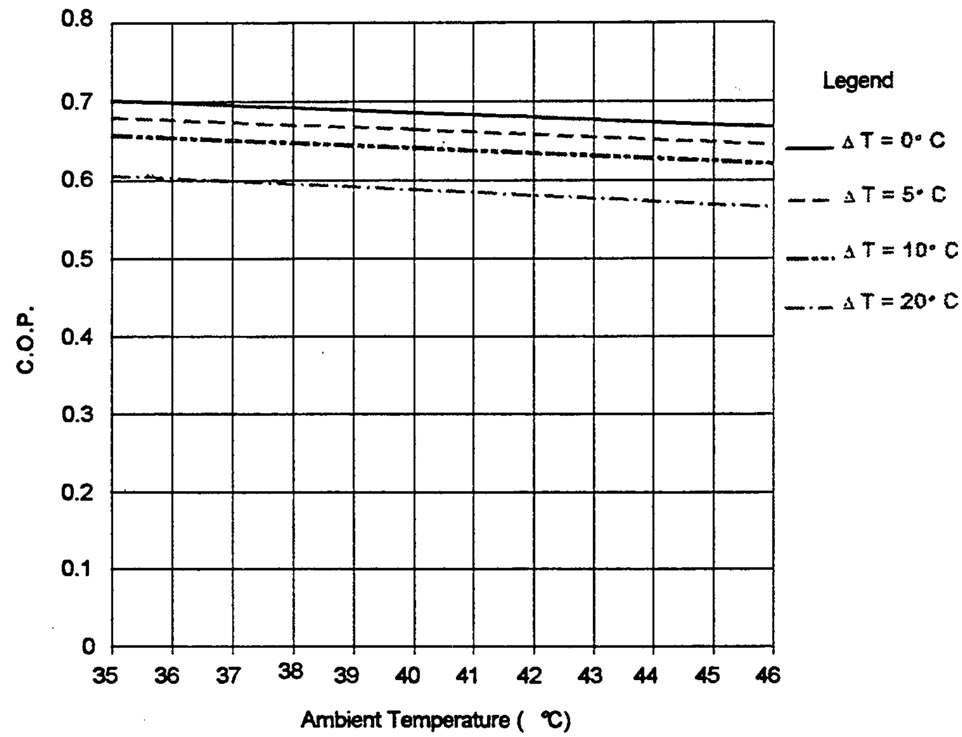

Figure F.3. Refrigeration COP. Curves of constant evaporator ΔT as a function of ambient temperature.

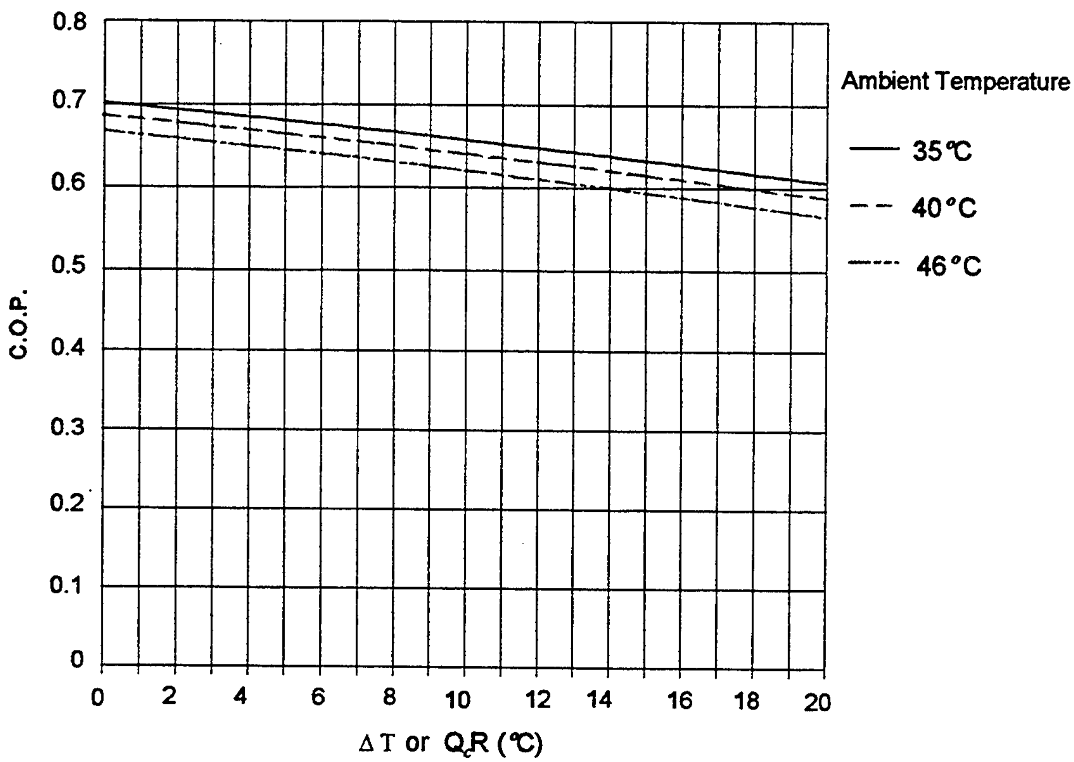

Figure F.4. Refrigeration COP. Curves of constant ambient temperature as a function of condenser ΔT.

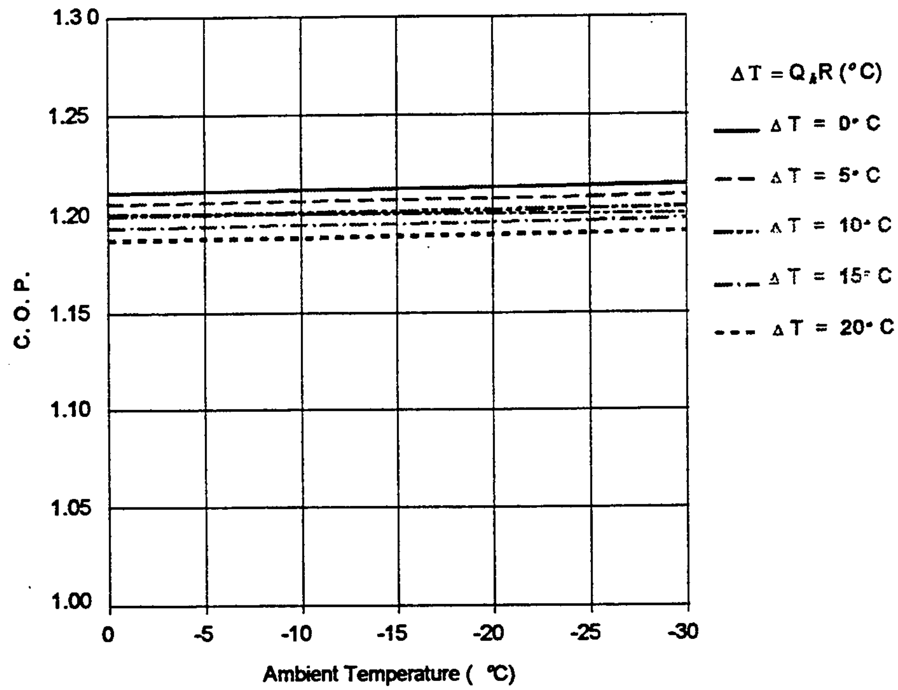

Figure F.5. Heating COP. Curves of constant condenser temperature ΔT as a function of ambient temperature.

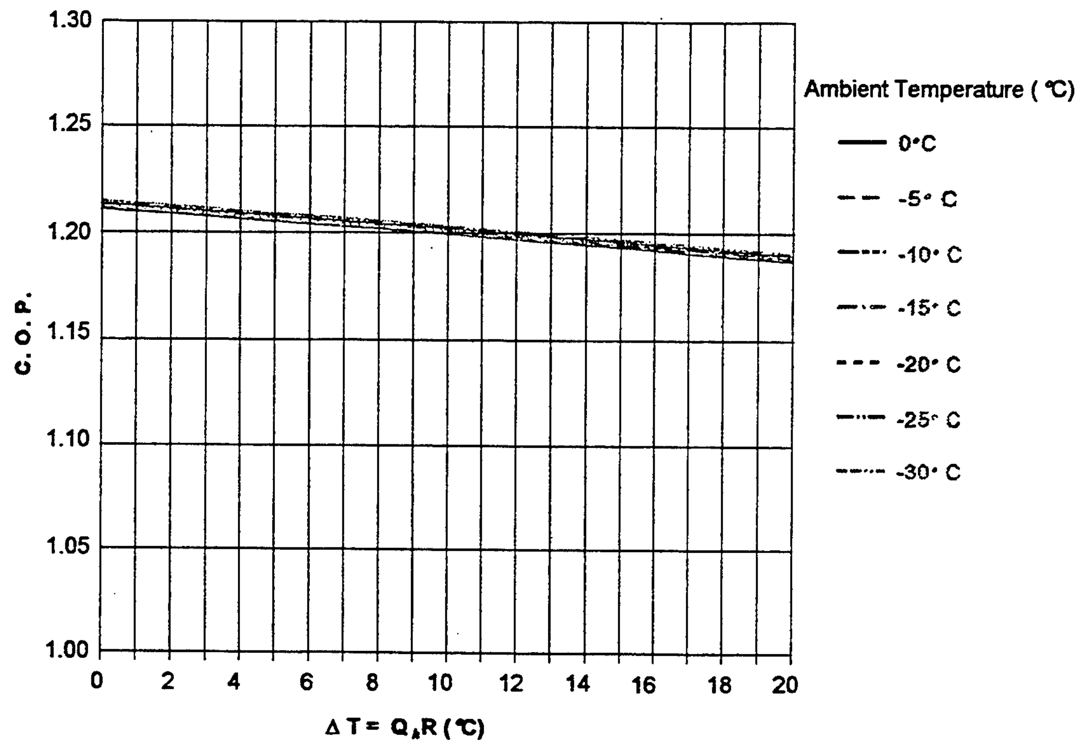

Figure F.6. Heating COP. Curves of constant ambient temperature as a function of condenser ΔT.

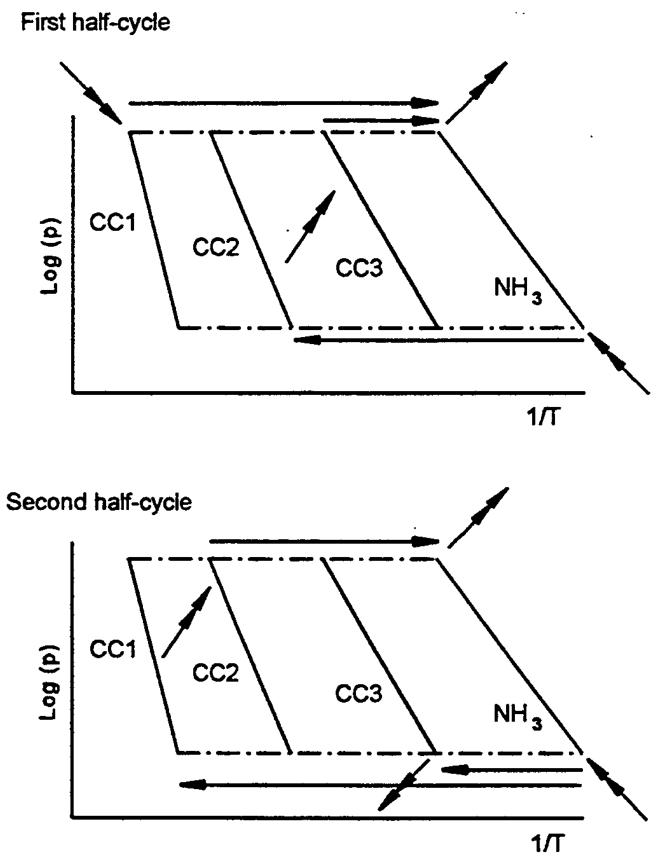

Figure F.7. Three stage cycle. Double arrows indicate heat flow, single arrows indicate ammonia flow.

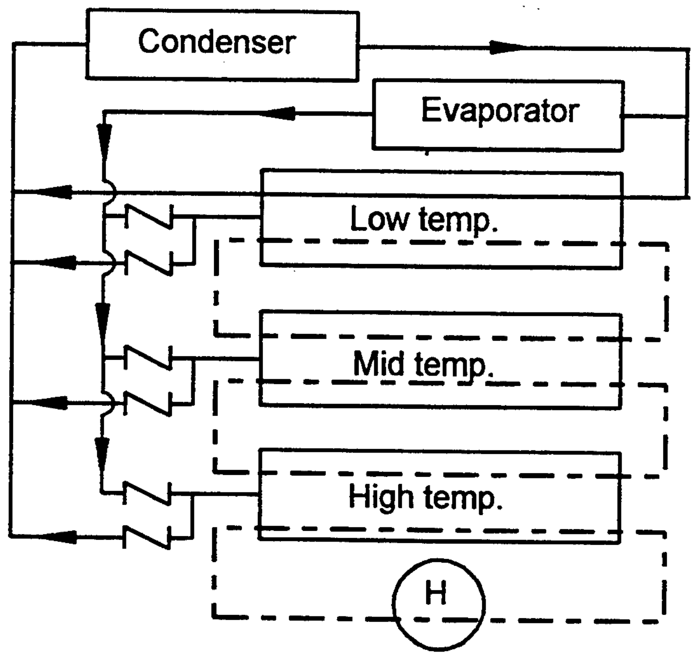

Figure F.8. Embodiment of a three stage heat pump. Dashed lines are heat transfer loops, solid lines are ammonia loops, valves are one-way, H is a heater.

G. Stirling Cycle Heat Pumps

G.1 General description.

A Stirling cycle heat pump is a combination of a Stirling engine and a Stirling refrigerator. Stirling cycle engines and refrigerators utilize external combustion and consist of fluid-containing working volumes separated by pistons and/or displacers, heat exchangers, and regenerators. They operate by the expansion and compression of a gaseous working fluid. Typically in a Stirling heat pump there is no external drive. In a heat pump the engine and refrigerator are connected by linkages which may be completely contained within the heat pump. Flow control of the working fluid in a Stirling device is accomplished by volume changes in the various chambers. Stirling's original engine contained a displacer as well as a piston. The displacer is a piston which divides a chamber into two parts and which has a very small pressure drop across it. Thus the gas temperatures in the two chambers differ, but their pressures are nearly the same. Some modern embodiments of Stirling devices use linkages to insure that the displacers are in proper phase with the pistons.

Stirling heat pumps can be classified into a number of different categories, depending on configuration of the chambers, linkages, and whether the total working gas volume is constant or varying during a cycle. Stirling heat pumps which have pressure changes resulting from changing of the bulk average fluid temperature by external heating while leaving the total volume constant are termed thermal compression heat pumps. If the pressure changes are due to varying both the total volume (through piston motion) and the fluid temperature through heat exchangers they are termed mechanical compression heat pumps. Generally, thermal compression engines need a larger working volume than mechanical compression heat pumps to achieve the same heating/cooling capacity. Thermal-compression devices use the power produced by the engine instantaneously in the refrigerator. Because the working volume is constant, by definition no net work can be produced.

Stirling heat pumps come in many different configurations. Figure G.1 shows one of the simplest of these, a piston-piston engine or refrigerator, no displacers, in what is called the alpha configuration. The pistons are mechanically linked, and two heat exchangers and a regenerator thermodynamically connect the two chambers. The gas flow is totally controlled by the volume changes, and the total volume occupied by the gas can vary over the cycle. Figure G.2 shows a Stirling engine or refrigerator with a piston and displacer in what is termed the beta configuration. The displacer is essentially a piston with no pressure drop across it. With the substitution of the displacer for a piston, the makeup of the device is similar to the alpha configuration.

An example of a mechanically coupled Stirling heat pump is shown in Figure G.3. This is essentially a pair of the beta configuration devices with the piston and displacer pairs each linked together.

A Stirling cycle heat pump separates the fluid in the engine section from the fluid in the refrigerator section. This separation in principle allows the engine and the refrigerator to operate at different pressures. Also, the warm temperature in the engine can be different from the warm temperature in the refrigerator. An adaptation of the Stirling heat pump is the Vuilleumier heat pump, which can be thought of as a Stirling heat pump where the refrigerator and engine compartments are allowed to communicate with one another. In that case the warm-hot compartment of the engine and the warm-cold compartment of the refrigerator are at very nearly the same temperature and pressure. A variation of this is the Stirling-Vuilleumier heat pump shown in Figure G.4. The difference between the figures is that the hot/warm and warm/cold spaces are linked so that gas can pass between them, equalizing pressure and temperature in these two spaces. This Vuilleumier adaptation of the traditional Stirling device keeps the total volume of the working space constant. A more elaborate variation of this is seen in Figure G.5, which is a simplified schematic of what has been termed a balanced-compounded Vuilleumier heat pump. Other variations are possible, one of which is shown in Figure G.6 in what was termed by the inventor as the Ericsson configuration. An Ericsson device is basically a Stirling or Vuilleumier device where the gas flow is controlled by valves. Such is not the case here, but the thermodynamic cycle is close to the traditional Ericsson cycle.

Another variation of the conventional Stirling refrigerator is the pulse-tube refrigerator, introduced for cooling near cryogenic temperatures. The original pulse-tube concept was the Stirling-based McMahon-Gifford refrigerator, which is basically a Stirling refrigerator but with the gas introduced in pulse-like fashion through valves. The pulse-tube refrigerator invented by Gifford and Longsworth [Gifford and Longsworth 1964, 1965] (Figure G.7). It has the advantage of no cold moving mechanical parts, and is capable of producing useful refrigeration at about 6.5 K in a multi-stage version. Gas at room temperature is introduced through the intake valve into the pulse tube, which is a thin-walled, poorly conducting tube closed at one end. During the filling process the entering gas is rapidly pulsed causing the gas already in the tube to be compressed. This causes a temperature stratification in the tube and the heat of compression is extracted by means of the hot heat exchanger. The exhaust valve is next opened, allowing the gas to expand and leave the tube. This expansion causes a cooling effect, and heat enters through the cold heat exchanger. The regenerator, usually a container filled with a stack of metal gauze plates or spheres, is used to cool the incoming gas. Pulse tubes are generally operated at pulse rates of 1 to 5 hertz and pressures of 2 to 10 bars. Efficiencies of these devices are sketchy, but calculations by Richardson for a pulse tube refrigerator operating between 17°C and -98°C show efficiencies to be low, at about 1% of the ideal Carnot efficiency.

Mikulin [Mikulin 1984] improved this refrigerator by adding an orifice and a large reservoir at the closed end (Figure G.8). In this version the gas is cooled by expanding through the orifice rather than by transferring heat with the pulse tube wall. This change greatly increases the enthalpy flow. Experiments [Radebaugh et al 1988, Storch and Radebaugh 1988] were conducted using a pulse tube with a 9.53 outer diameter, 9 inner diameter, and 110 millimeter length. The regenerator was 15.9 millimeter outer diameter, 15.3 millimeter inner diameter, 100 millimeter long, and filled with 200 mesh stainless steel screen with a porosity of 0.63. The swept volume of the compressor was 25 cubic centimeter and the reservoir volume was 500 cubic centimeter. The base pressure was 1,500 kilopascals and the pulsing frequency was 15 hertz. They found that the power output versus cooling temperature was approximately

$$\text{Refrigeration power in watts} = 0.0857*[T(^{\circ}C) + 200]. \quad (G.1)$$

No efficiencies were given. Other investigations have also been carried out [Richardson 1983, Wang et al 1992].

Another way in which Stirling devices can differ is in the mechanical drive mechanism. Classic drive systems use either a slider-crank drive as in IC engines, a swashplate drive, the latter imparting sinusoidal motion to the pistons and providing a degree of engine control, or a scotch yoke [Reader and Cooper 1983, Urielli and Berchowitz 1984, Wurm et al 1991, Organ 1992, Walker et al 1994]. These drives have the severe disadvantage of applying a high side load on the piston, with resulting drag and wear due to friction on piston and cylinder. A rhombic drive (Figure G.9) was developed at N. V. Philips [Meijer 1959] to seal off the cylinders from the crankcase and to provide dynamic balance even for a one-cylinder engine. A yoke device developed by Ross is another drive mechanism which has become increasingly popular (Figure G.10).

A key element in the operation of both the Stirling engine and the refrigerator is the regenerator (originally called an economizer by Stirling), a form of heat exchanger which acts as a thermal storage device. As gas passes from a hot space to a colder space, it passes through the regenerator which absorbs thermal energy from the working fluid. When the cool air is returned to the hot chamber, the regenerator releases the stored heat, thereby increasing the thermal efficiency of the engine.

During portions of a cycle mechanical compression heat pumps produce either a surplus or a deficit of the work needed by the refrigerator. Piston inertia is used to store sufficient energy during the surplus period to provide for the deficit period later in the cycle.

Mechanical compression Stirling cycle heat pumps have been constructed in a free piston arrangement so that no mechanical linkages are necessary. The gas acts as a spring, moving the piston from one end of the cylinder to the other. The lack of an external drive makes it easier to realize a hermetically sealed heat pump for the mechanical-compression concept, especially with the free piston arrangement. Free piston heat pumps may require an active control system to insure that the gas spring is tuned to the piston mass so as to reduce vibration and noise and to ensure proper phasing of the piston.

Stirling heat pumps operate with working fluids which are always in the gaseous state. Helium and nitrogen are frequently used because of their low molecular weights which results in increased efficiencies. However both helium and nitrogen are difficult to contain and some leakage is inevitable, requiring periodic replenishment. Other gases can be used, with an attendant increase in size and decrease in efficiency. Air has many attributes which make it the working fluid of choice in many cases. It is necessary to only provide a small compressor to maintain the heat pump against any leakages, and there are no hazardous factors associated with air. Generally, at higher engine speeds (above 1,200 rpm or so) the use of hydrogen for the working fluid gives the highest energy density (power output per unit volume of engine), with helium having an energy density roughly 20% lower and air about 75% lower than hydrogen. At lower engine speeds (below 1,000 rpm or so) air may actually outperform both helium and hydrogen.

Any heat source can be used to drive the engine, including electric resistance and solar heating. A thermally driven Stirling heat pump has all linkages contained within the case, so that operation is relatively noise free compared to internal combustion engines. Stirling refrigerators can be driven by motors through mechanical linkages rather than by Stirling engines. Performance and fuel economy of the Stirling engine is comparable to a diesel engine, with the Stirling engine having superior emission characteristics. The cost of a Stirling engine has been estimated as being two to three times that of a diesel engine, with the cost penalty being higher for smaller duty engines.

Stirling engines and refrigerators have been designed to run over the following ranges of parameters:

hot-end temperatures (engine) - 50°C (120°F) to over 750°C (1,380°F)
cold-end temperatures (engine) - typically above 0°C (32°F) to allow water cooling
mean pressures - 0.1 MPa (1 bar, 14.7 psia) to 15 MPa (150 bars, 2,200 psia)
operating speeds - 750 to 3600 rpm

Some back-of-the-envelope formulae have been developed for quick estimates of operating parameters based on observations by W. Beale of Sunpower Inc. One commonly used form for power output of a Stirling engine is

$$W_E = 0.015\, p\, f\, V \qquad \text{(G.2)}$$

where

W_E = engine power output in watts,
p = mean cycle pressure in bars (psia divided by 14.7),
f = operating frequency in hertz (cycles per second),
V = swept volume in the compression space in cubic centimeters.

This equation is particularly useful for Stirling engines running at hot temperatures around 1000 K (727°C or 1,340°F) and low temperatures around 300 K (27°C or 80°F). To account for other temperature ranges this equation can be stated in terms of temperatures according to

$$W_E = 0.025\, p\, f\, V\, (T_H - T_C) / (T_H + T_C), \qquad \text{(G.3)}$$

where T_H and T_C are the hot and cold temperatures, respectively.

For the refrigerator a formula corresponding to (G.3) is

$$W_R = 10^{-6} p\, V f\, T_R \qquad \text{(G.4)}$$

with

W_R = refrigeration capacity in watts,
T_R = cold expansion space temperature in degrees Kelvin.
Other quantities defined as for the engine.

For the Stirling engine, thermal efficiencies between 15% and 35% have been obtained. The higher end of this efficiency range requires a high degree of technological expertise by the designer. For refrigerators, performance is usually rated in terms of the coefficient of performance (COP), defined as

$$COP_{Refrigeration} = \text{heat lifted/work done.} \qquad \text{(G.5)}$$

For a Stirling refrigerator a typical range of values for the refrigeration COP which have been obtained are given by

$$COP_{Refrigeration} = c\, T_R / (T_C - T_R), \qquad \text{(G.6)}$$

where

T_C = cooled (ambient) space temperature,
T_R = cold (refrigerated) space temperature,

and c is a number varying from 0.01 to 0.45. The lower part of this range for c applies to very small cryocoolers and the upper part to very large cryocoolers.

The thermodynamic cycle of an actual Stirling heat pump is difficult to describe, because of factors like friction, piston leakage, and heat exchanger inefficiencies which enter into the actual construction of the device. Simple models which have been developed are the ideal isothermal model (pressure-volume and temperature-entropy curves shown in Figures G.11 and G.12), the adiabatic model (pressure-volume and temperature-entropy curves shown in Figures G.13 and G.14), and the quasi-steady model (pressure-volume and temperature-entropy curves shown in Figures G.15 and G.16), each model being successively more sophisticated and realistic than the previous one. The computer models used and developed under this contract use all three of these models, the simpler models being used to obtain a general picture of the size and capabilities of the heat pump and the more elaborate model to give a more realistic picture of the output and losses. Results from these calculations are shown in Figures G.17 through G.22. In all of these figures the room temperature is set at 22°C (71.6°F) and the gas is hydrogen.

Figures G.17 and G.18 show refrigeration coefficient of performance (COP) as a function of either ambient temperature or ΔT, the difference between the temperature of the gas inside the heat exchanger and the temperature from which heat is being absorbed. The highest COP is obtained when this temperature difference is zero, the "ideal" heat exchanger. Thus the ΔT curves are a function of the heat exchanger efficiency, defined as a thermal resistance according to

$$R_{thermal} = \Delta T/Q. \qquad \text{(G.6)}$$

Figures G.19 and G.20 are corresponding results for heating.

In Figures G.17 through G.20 the regenerator is idealized by assuming that no pressure drop occurs across it. Since the regenerator is just a container filled with a porous media, in reality the regenerator pressure drop can be an important factor in determining heat pump efficiency. Figures G.21 and G.22 are for the same operating conditions as Figures G.17 and G.18, but with a realistic regenerator pressure drop included. It is seen that the effect of the regenerator pressure drop is to reduce the COP by a factor of approximately 2. The engineering difficulties involved in improving piston/cylinder wear, heat exchangers, and regenerator performance have been among the chief factors in holding back the applications of Stirling technology.

Figure G.23 shows the differences in performance of a Stirling heat pump for different gases at an operating speed of 1,500 rpm. Hydrogen has the highest efficiency, with air second and helium third. Room temperature is the same as in Figures G.17 through G.22, and the regenerator pressure drop is taken as zero. The ranking of the various gases as to operating efficiency will vary with operating conditions and engine speed. These results are representative of Stirling behavior at speeds above 1,200 rpm. Below 1,200 rpm the relative ranking of the gases change.

G.2 Advantages.

a. Stirling devices can use virtually any heat energy source.

b. Stirling heat pumps operate quietly.

c. They have minimal pollutant emissions, using no CFCs or HCFCs. With air as the working fluid, combustibility and toxicity hazards to personnel is virtually nil.

d. Stirling heat pumps have a low consumption of lubricants and low internal wear.

e. They have a low exhaust temperature, reducing personnel hazards and heat signature.

f. Water can be used as a lubricant for these heat pumps in some temperature ranges.

G.3 Disadvantages.

a. Manufacturing costs are high. Cost estimates of about twice that of a diesel engine are often quoted.

b. Seal reliability has been a problem, requiring special designs. An advantage of the piston-displacer design is that the diameter of the displacer rod is much less than the diameter of the piston, with proportionately less leakage, friction, and seal size.

c. Heat exchangers generally are quite large in size.

d. The necessary control system can be quite complicated. Temperatures, pressures, stroke, phase angle, dead volume, speed, and applied heat all must be controlled to insure proper operation. Response time of Stirling engines is relatively slow because of large thermal inertias.

e. Operating temperatures on the hot side of the engine generally are quite large, typically in the 1000 K range, giving high thermal signatures, possible fire hazards, and also posing a fire hazard to nearby combustible materials such as buildings and vehicles.

f. Except for a free piston design, Stirling engines are not self-starting, and so a starting motor is a necessary part of the design. This adds to the cost and size.

g. Design equations of the form (1) above show that to increase power output it is necessary to either operate at high pressure levels or to have large volumes. The former presents potential hazards to personnel, while the latter affect both the number of ECUs that can be transported on a pallet and the time and number of personnel needed for setup.

G.4 Tradeoffs, Hazards, and Risks Analysis.

The Stirling engine was born in 1816 and enjoyed some popularity in the 19th century, although many of the devices did not use a regenerator. They fell out of favor with the introduction of the modern internal combustion and diesel engines. Meijer at N V. Philips in The Netherlands was responsible for renewed interest in the technology, and introduced many of the concepts still in vogue today. There have been many efforts by the automotive industry and government agencies to advance the technology, but in the past something has always happened to stop the projects before they were brought to fruition. Many advances have been made in the technology since the 1950's, and a number of foreign countries have pursued intense government-supported development activities, yet there have been few applications where Stirling devices have been able to compete with older technologies. In the heat pump market, the ECU is in the range of home heating/cooling, where first cost tends to dominate the market. There presently appears to be little activity in pursuing this market at the present time, as costs, size, and complexity are orders of magnitude above what is presently available.

The tradeoff between size and gas pressure is a strong limiting factor for the ECU needs of the Air Force. This tradeoff is worsened if a benign gas such as air is used for the working fluid, rather than the more thermally efficient nitrogen or helium, for even greater volume is necessary. To meet the Air Force requirement of stacking on pallets it is necessary to utilize higher pressures, increasing both weight requirements and the potentiality for accidents. At the levels of pressure necessary, a pinhole leak would result in a gas jet which could be dangerous to operators. Temperature levels are also high, resulting in further operator hazards, a potential combustion source, and a thermal signature that would be difficult to disguise. Control complexity would require special training of operator/maintenance personnel, and setup time could be lengthened by the machinery needed to move the weight of the device.

G.5 Manufacturers.

There are presently no manufacturers who provide an "off-the-shelf" design. The following are actively engaged in the development and design of special application Stirling devices.

Mechanical Technology, Inc., 968 Albany Shaker Rd., Latham, NY, 518-785-2299.

Stirling Machine World, 1823 Hummingbird Ct., West Richland WA

Stirling Power Systems Corp., 275 Mety, Scio Twnsp., Ann Arbor MI 48014, 313-665-6767.

Stirling Thermal Motors Inc., 2841 Boardwalk, Ann Arbor MI 48014, 313-995-1755.

Sunpower Inc., 6 Byard St., Athens OH, 614-594-2221.

United Stirling Inc., Alexandria VA.

G.6 References.

Gifford, W. E., and R. C. Longsworth, " Pulse tube refrigeration", Trans. of ASME vol. **63**, page 264, 1964.

Gifford, W. E., and R. C. Longsworth, " Pulse tube refrigeration progress", Adv. Cryogenic Engineering vol. **10**, pages 69-79, 1965.

McMahon, H. O., and W. E. Gifford, "A new low-temperature gas expansion cycle", Adv. Cryogenic Engineering vol. **5**, pages 354-372, 1960.

Meijer, R. J., "The Philips hot-gas engine with rhombic drive mechanism", Philips Technical Review vol. **20**, pp. 245-276, 1959.

Mikulin, E. I., A. A. Tarasov, and M. P. Shkrebyonock, "Low-temperature expansion pulse tube", Adv. Cryogenic Engineering vol. **29**, page 629, 1984.

Organ, A., *Thermodynamics and Gas Dynamics of the Stirling Cycle Machine,* Cambridge University Press, Cambridge, 1992.

Radebaugh, R., K. Chaudry, and J. Zimmerman, "Optimization of a pulse tube refrigerator for a fixed compressor swept volume", Proc. Fifth Internat. Cryocooler Conf., Naval Postgraduate School, Monterey CA, page 133, 1988.

Reader, G., & C. Cooper, *Stirling Engines*, E. & F. N. Spon, New York, 1983.

Richardson, R. N., "Pulse tube refrigeration - an investigation pertinent to cryocooler development", D. Phil. Thesis, Oxford University, 1983.

Storch, P. J., and R. Radebaugh, "Development and experiment test of an analytical model of the orifice pulse tube refrigerator", Adv. Cryogenic Engineering vol. **33**, Plenum Press, New York, pages 851-859, 1988.

Urieli, I, & D. M. Berchowitz, *Stirling Cycle Engine Analysis*, Adam Hilger, Bristol, 1984.

Walker, G., G. Reader, O. R. Fauvel, E. R. Bingham, *The Stirling Alternative Power Systems, Refrigerants and Heat Pumps*, Gordon & Breach Science, Yverdon, 1994.

Wang, C., P. Wu, and Z. Chen, " Numerical modelling of an orifice pulse tube refrigerator", Cryogenics vol. **32**, pages 785-790, 1992.

Wurm, J., J. A. Kinast, T. R. Roose, & W. R. Staats, *Stirling and Vuilleumier Heat Pumps*, McGraw-Hill 1991.

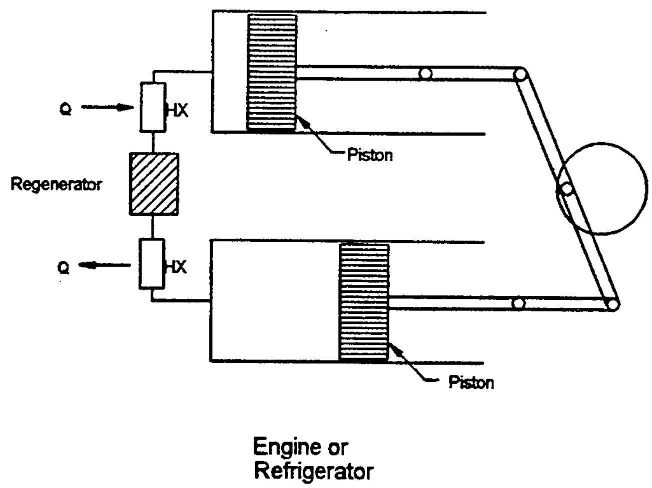

Figure G.1. Stirling engine or refrigerator schematic in piston-piston (alpha) configuration.

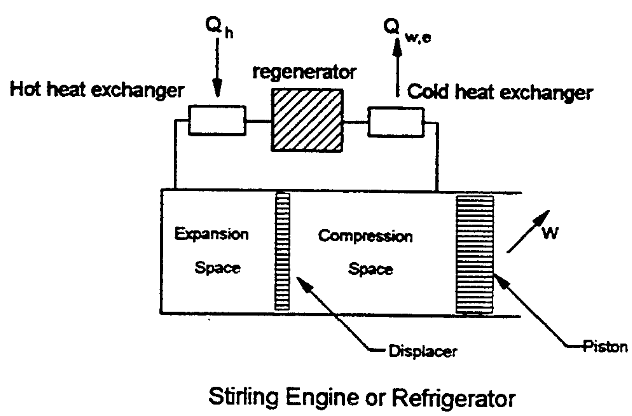

Figure G.2. Stirling engine or refrigerator in displacer-piston (beta) configuration.

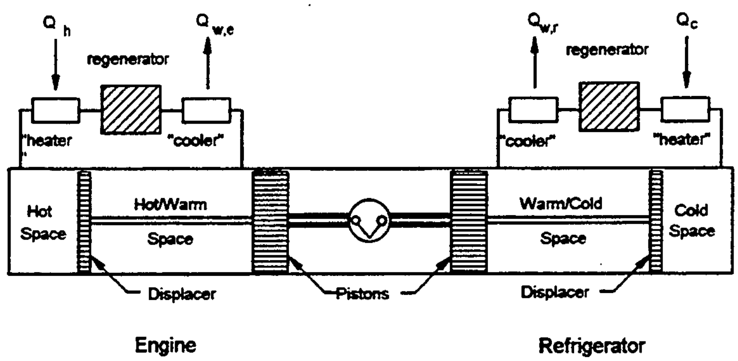

Figure G.3. Schematic of mechanically coupled mechanical compression thermally driven Stirling heat pump in piston-displacer configuration.

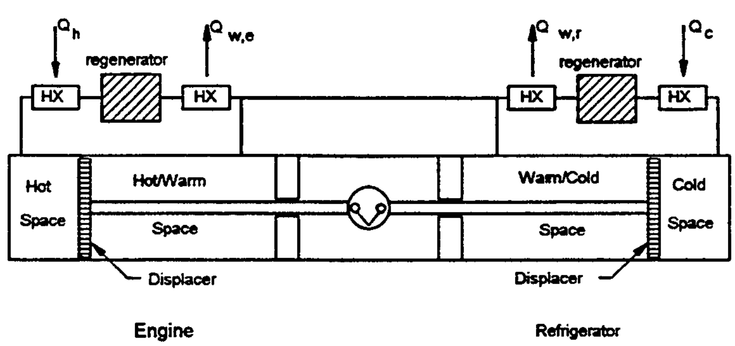

Figure G.4. Schematic of mechanically coupled thermal compression thermally driven Stirling-Vuilleumier heat pump.

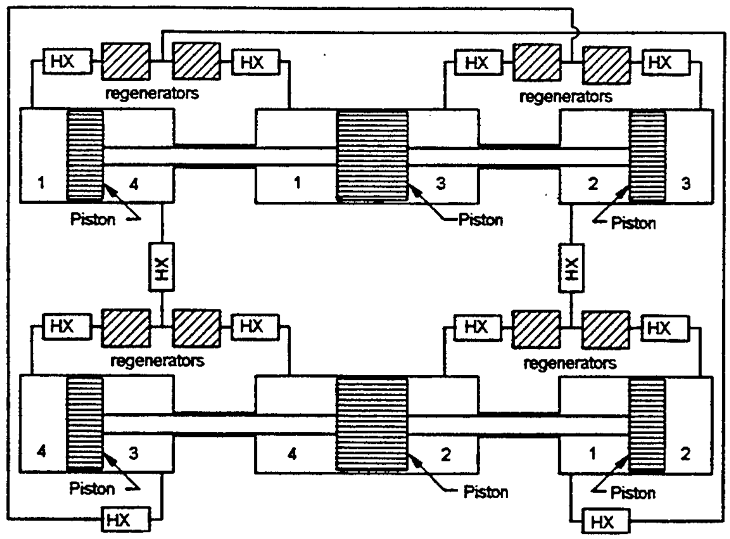

Figure G.5. Mechanical compression Stirling-Vuilleumier heat pump.

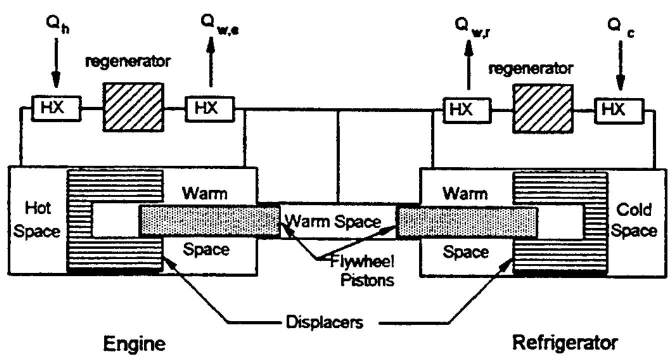

Figure G.6. Stirling heat pump - Ericsson configuration.

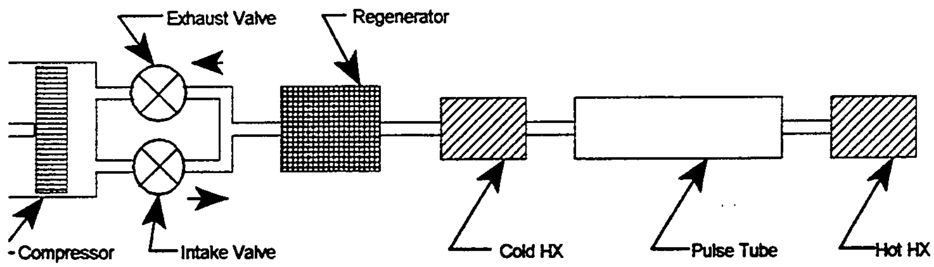

Figure G.7. Gifford-Longsworth valve-controlled single-stage pulse tube refrigerator.

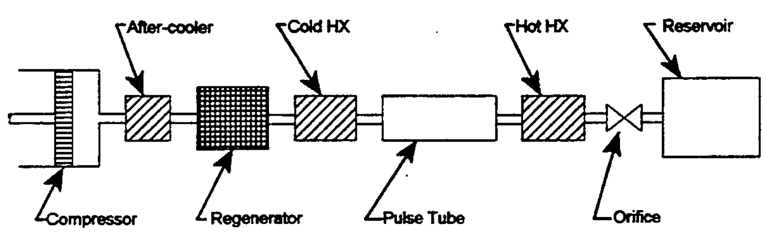

Figure G.8. Valve-less orifice pulse tube refrigerator.

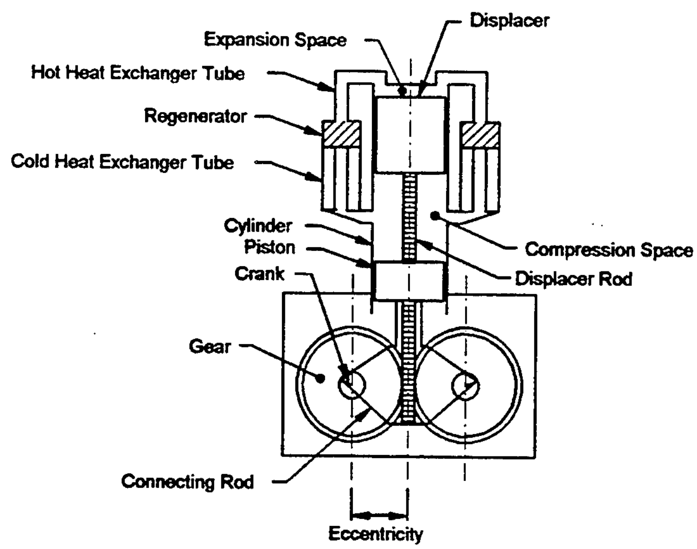

Figure G.9. Example of Stirling Ross rhombic drive.

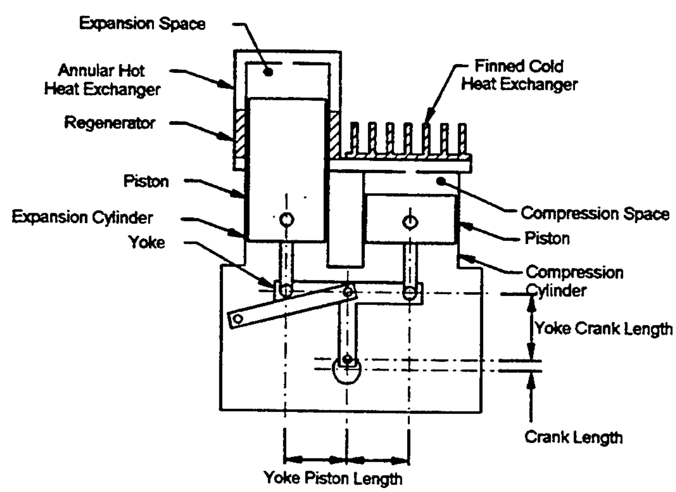

Figure G.10. Example of Stirling yoke drive.

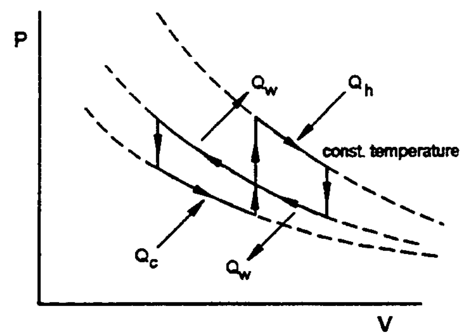

Figure G.11. Ideal isothermal Stirling cycle heat pump pressure-volume diagram.

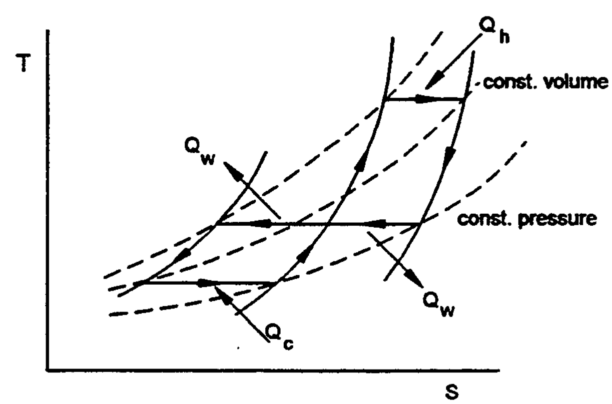

Figure G.12. Ideal isothermal Stirling cycle heat pump temperature-entropy diagram.

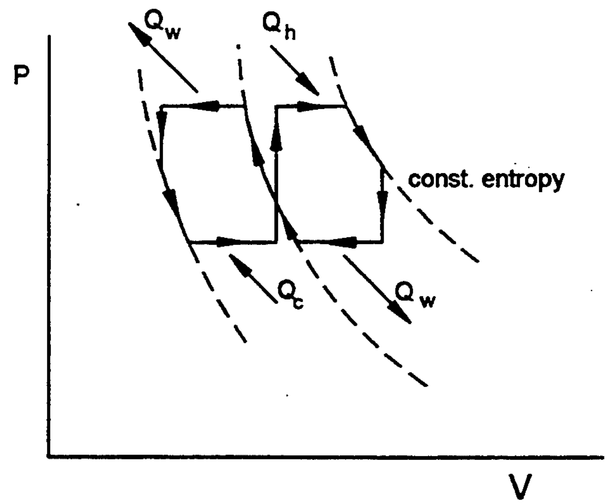

Figure G.13. Ideal adiabatic Stirling cycle heat pump pressure-volume diagram.

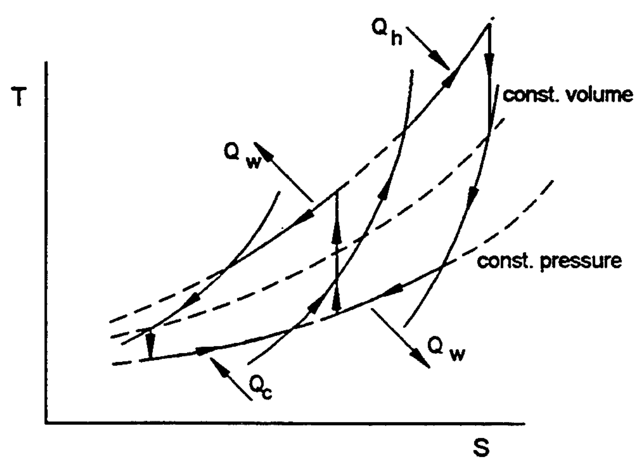

Figure G.14. Ideal adiabatic Stirling cycle heat pump temperature-entropy diagram.

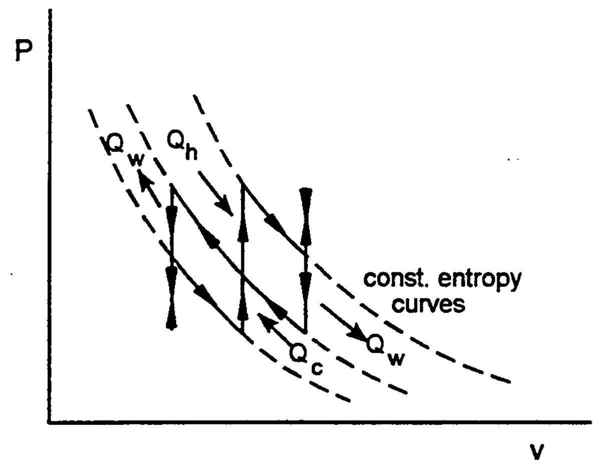

Figure G.15. Stirling cycle heat pump pressure-volume diagram for quasi-steady analysis.

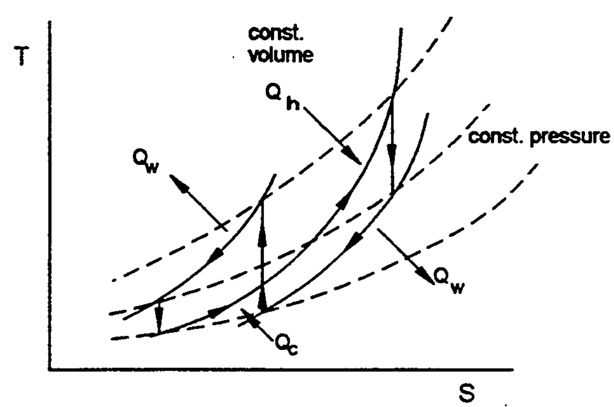

Figure G.16. Stirling cycle heat pump temperature-entropy diagram for quasi-steady analysis.

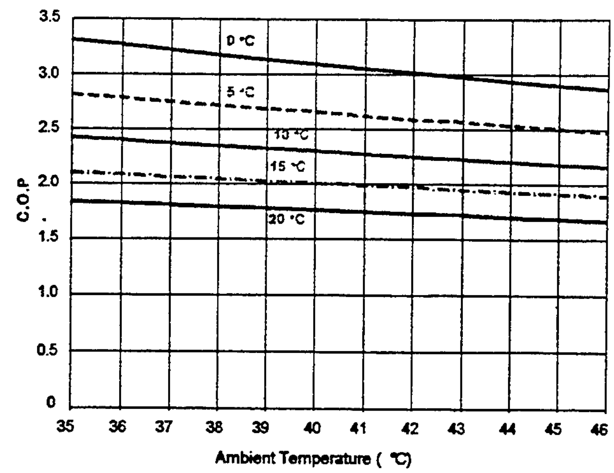

Figure G.17. Refrigeration COP as a function of ambient temperature. Curves of constant ΔT. Gas is hydrogen.

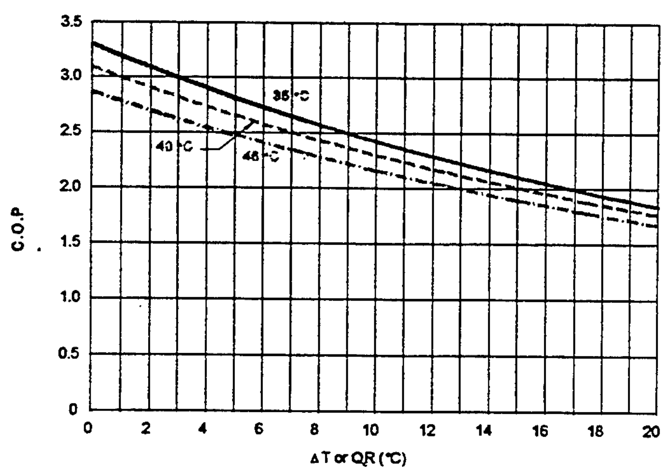

Figure G.18. Refrigeration COP as a function of ΔT. Curves of constant ambient temperature. Gas is hydrogen.

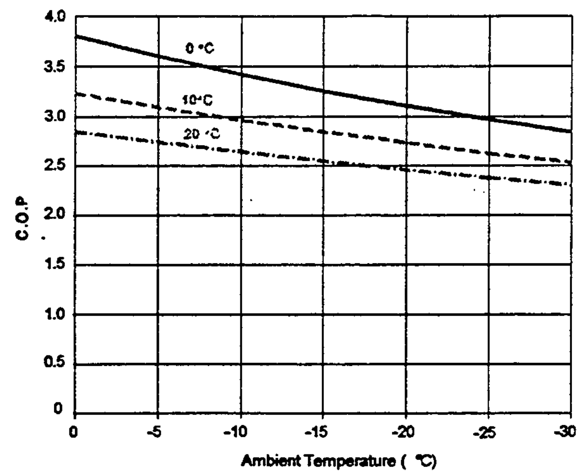

Figure G.19. Heating COP as a function of ambient temperature. Curves of constant ΔT. Gas is hydrogen.

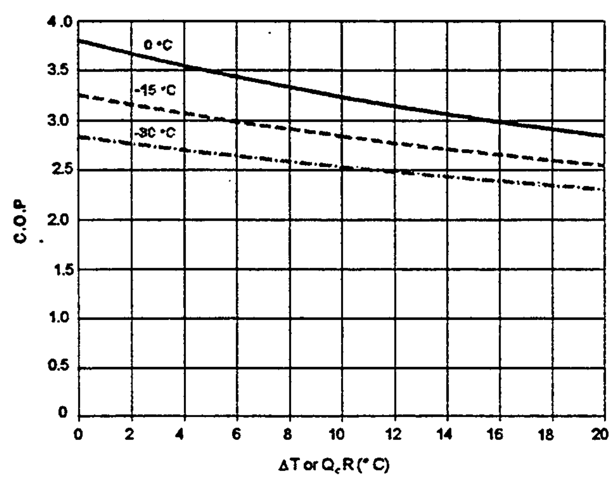

Figure G.20. Heating COP as a function of ΔT. Curves of constant ambient temperature. Gas is hydrogen.

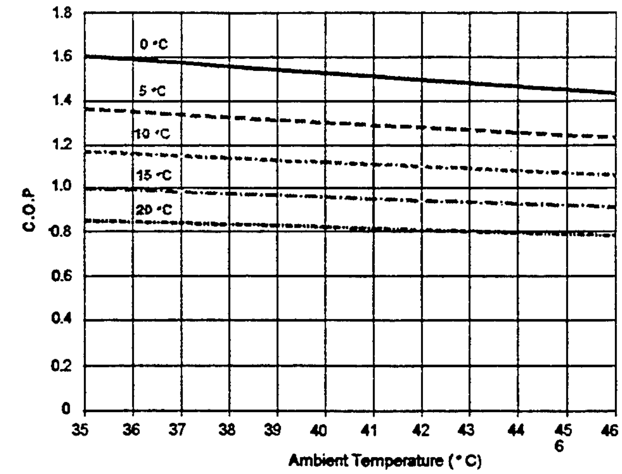

Figure G.21. Refrigeration COP as a function of ambient temperature. Curves of constant ΔT. gas is hydrogen.

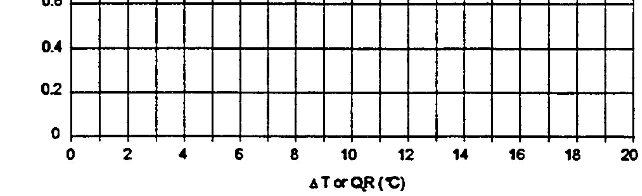

Figure G.22. Refrigeration COP as a function of ΔT. Curves of constant ambient temperature. Gas is hydrogen.

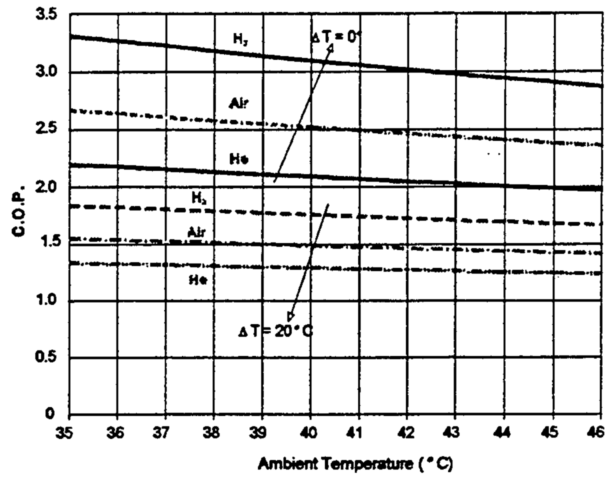

Figure G.23. Comparison of cooling efficiencies of three gases in a Stirling refrigerator. Operating speed 1,500 rpm.

H. Thermoelectric Heat Pumps

H.1 Thermoelectric Effects

Thermoelectric devices can be used for electricity generation as well as heating and cooling. They are based on the same physical phenomenon as the familiar bimetal thermocouple used for temperature measurement, but for heat pumping applications semiconductor materials provide much higher efficiencies than the traditional copper-constantan and similar thermocouples used for measurement purposes.

The phenomenon of heat pumping by a thermoelectric device depends on several physical phenomena. The Peltier effect produces the actual heat pumping effect, while Fourier heat transfer and Joule electric resistance heating are dissipative effects which limit the range of utilization of the heat pumping. All of these processes were understood in the early nineteenth century, but efficient application of the Peltier effect was not possible until the properties of modern semiconductors were available. General references are [Bridgeman 1954, Rosi et al 1959, Cadoff and Miller 1960, Egli 1960, Kaye and Welsh 1960, MacDonald 1962, Chang 1963, Soo 1968, Angrist 1976]. References specific to cooling are [Penrod 1959, Eichhorn 1963].

In the following discussion, p denotes a p-doped semiconductor and n denotes an n-doped semiconductor.

H.1.1 Seebeck and Peltier Effects (Reversible).

The Seebeck effect is the phenomenon used in temperature measurement by thermocouples. When two dissimilar conductors p and n are connected so that the junctions are at different temperatures a potential difference ΔV is found to have been developed across the conductors. Denoting the temperature difference across the junctions by ΔT, the potential difference is given by

$$\Delta V = \alpha_{pn}\Delta T, \text{ where } \alpha_{pn} = \alpha_p - \alpha_n. \qquad \text{(H.1)}$$

The coefficient α_{pn} is referred to as the differential material Seebeck coefficient.

The Peltier effect states that when two dissimilar conductors p and n carry an electric current I, heat is taken in at one of the junctions and given off at the other junction. If the current goes from the p material to the n material, the junction of p and n is cooled. Reversing the current heats this junction. The rate of heat flow is given by

$$Q_{pn} = \pi_{pn} I, \text{ where } \pi_{pn} = \pi_p - \pi_n \qquad \text{(H.2)}$$

and Q_{pn} is the rate of heat transfer at the junction of p and n. The parameter π_{pn} is referred to as the differential material Peltier coefficient.

The net result in using these principles either in heat pumping or electricity generation can be combined into one simple equation at a junction, namely that the rate of heat flow at a junction of a p and an n semiconductor is given by

$$Q = \alpha_{pn} \, I \, T, \qquad \text{(H.3)}$$

where T is the temperature of the junction.

In a given material the Seebeck and Peltier coefficients α_{pn} and π_{pn} depend strongly on temperature. These effects are both reversible in the sense that changing the direction of current flow changes the direction of heat flow.

H.1.2 Fourier and Joule Effects (Irreversible).

Traditional heating effects which act to reduce thermoelectric heat pumping and electrical generation are heat conduction and electric heating. The well known Fourier law of heat conduction states that heat transfer occurs in the presence of a temperature difference occurring over a length L according to

$$Q = kA\Delta T/L, \qquad \text{(H.4)}$$

where k is the thermal conductivity of the material. The factor kA/L represents the thermal resistance of the material.

Joule heating occurs due to the electrical resistance of a material. The rate of generation of heat is given in a material of cross-sectional area A and length L by

$$Q = \rho L I^2/A, \qquad \text{(H.5)}$$

where ρ is the electrical resistivity of the material.

The thermal conductivity and electrical resistivity depend on temperature. Changing current direction has no effect on the direction of heat flow.

H.2 Thermoelectric Heat Pumping.

A thermoelectric module consists of a number (N) of semiconductor junction pairs. These pairs are usually connected in series electrically and in parallel thermally. For a module consisting of N thermocouple pairs the assembly of thermocouples is seen as a unit, and equations (H.3) to (H.5) are replaced by

$$Q_{PeltierCold} = N\alpha_{pn} I T_{Cold} = S_M \, I \, T_{Cold}, \qquad \text{(H.6)}$$

$$Q_{PeltierHot} = N\alpha_{pn} I T_{Hot} = S_M \, I \, T_{Hot}, \qquad \text{(H.7)}$$

$$Q_{Fourier} = N(k_p A_p/L_p + k_n A_n/L_n)(T_{Hot} - T_{Cold}) = K_M (T_{Hot} - T_{Cold}), \qquad \text{(H.8)}$$

$$Q_{Joule} = N(\rho_p L_p/A_p + \rho_n L_n/A_n)I^2 = R_M I^2, \tag{H.9}$$

where S_M, K_M, and R_M are the module Seebeck, thermal conductance, and electrical resistance coefficients.

A thermodynamic analysis of such a module can be easily carried out. The above-described thermoelectric effects act on both the hot and cold side of a thermoelectric module. With T_{Hot} greater than T_{Cold}, the following notation is introduced:

Q_{Hot} = rate of heat rejection out of hot end,

Q_{Cold} = rate of heat addition into cold end.

Heat balances can now be taken on both the hot and cold side of a thermoelectric module. The Joule heating within a semiconductor leg flows equally into the hot and the cold junctions. The Fourier heat conduction flows out of the cold junction and into the hot junction. The heat flows at the hot and cold ends of the semiconductors are shown in Figure H.1 and Table H.1.

Table H.1. Heat flow terms at a thermoelectric junction

	Rate of heat flow out	Rate of heat flow in
Hot junction	$Q_{Hot} + Q_{Fourier}$	$Q_{PeltierHot} + 0.5Q_{Joule}$
Cold junction	$Q_{PeltierCold}$	$Q_{Cold} + 0.5Q_{Joule} + Q_{Fourier}$

A heat balance at each junction then leads to the following equations:

$$Q_{Hot} + Q_{Fourier} = Q_{PeltierHot} + 0.5Q_{Joule}, \tag{H.10}$$

$$Q_{Cold} + 0.5Q_{Joule} + Q_{Fourier} = Q_{PeltierCold}, \tag{H.11}$$

where the corresponding heat fluxes (all temperatures in the following are in degrees absolute, i.e. Kelvin or Rankine) are given by equations (H.6) through (H.9) plus

$$Q_{PeltierHot} = S_M\, I\, T_{Hot}, \tag{H.12}$$

Putting equations (H.6), (H.7), (H.8), (H.9) and (H.12) into equations (H.10) and (H.11) gives

$$Q_{Hot} = 0.5\, R_M\, I^2 + S_M\, I\, T_{Hot} - K_M\,(T_{Hot} - T_{Cold}) \tag{H.13}$$

= rate of heat rejection at the hot junction,

$$Q_{Cold} = -0.5\,R_M\,I^2 + S_M\,I\,T_{Cold} - K_M\,(T_{Hot} - T_{Cold}) \qquad (H.14)$$

= rate of heat intake at the cold junction,

$$W_L = Q_{Hot} - Q_{Cold} = VI = IS_M(T_{Hot} - T_{Cold}) + I^2R_M \qquad (H.15)$$

= power needed for heat pumping,

$$V_{Seebeck} = S_M(T_{Hot} - T_{Cold}) = \text{Seebeck voltage generated,} \qquad (H.16)$$

$$V = S_M(T_{Hot} - T_{Cold}) + R_M I \qquad (H.17)$$

= voltage required in a heat pump or

produced in an electricity generator,

$$\text{Cold coefficient of performance} = COP_{cold} = Q_{Cold}/W, \qquad (H.18)$$

It can be seen from equation (H.13) for instance that as the current increases the rate of heat pumping due to the Peltier effect increases linearly with the current, but the Joule heating increases as the square of the current. There is then an optimum current at which the heat pumping is a maximum, and if the current is increased to too high a value all cooling effect is lost. The optimum current which maximizes Q_{Cold} is

$$I_{optimum} = S_M T_{Cold}/R_M \qquad (H.19)$$

and the maximum current for heat pumping is

$$I_{maximum} = S_M T_{Cold}/R_M + \sqrt{(S_M T_{Cold}/R_M)^2 - 2K_M(T_{Hot} - T_{cold})/R_M} \qquad (H.20)$$

The effect of the semiconductor material properties on module performance can be further illustrated by introducing the material parameter

$$Z = S_M^2/R_M K_M. \qquad (H.21)$$

Rewriting equation (H.14) in terms of Z and the current ratio $i = I/I_{optimum}$, the result is

$$Q_{Cold} = K_M T_{Cold}\,[ZT_{Cold}(-0.5i^2 + i) + 1 - T_{Hot}/T_{Cold}]. \qquad (H.22)$$

The higher the value of ZT_{Cold} in the temperature range of interest, the higher will be the heat pumping capacity of the module.

The parameter Z depends only on the physical properties of the material and is commonly referred to as the figure of merit of the material. It has dimensions of reciprocal temperature. Occasionally Z' = TZ, a dimensionless quantity, is used as a figure of merit.

The Seebeck, Fourier, and Joule coefficients all can be satisfactorily approximated by polynomials, usually of either second or third order. A typical plot of Z versus temperature is parabolic opening downward. In selecting a semiconductor material it is important that the material used have its maximum Z very near the hot and cold temperatures of interest, for the rolloff from the maximum value is reasonably fast. For this reason bismuth telluride (Bi_2Te_3) doped usually either with antimony or selenium or a combination of both is most often used for applications near normal room temperature. It has a peak Z of approximately 0.0029 per degree C at about 70°C. Other materials may have a higher maximum Z at higher or lower temperatures, but their Z at room temperature is substantially lower than that of bismuth telluride. Today there are intensive investigations under way to find new ways of creating materials with improved figures of merit so as to improve the efficiencies of thermoelectric devices.

H.3 Multi-Stage Cascaded Modules.

The maximum effective temperature differential across a single-layer thermoelectric module is about 60°C (108°F). At that temperature differential the heat pumping capacity is nil. To increase heat pumping capacities, it is possible to stack the modules one on top of the other, cold side to hot side, so that they are thermally in series but electrically in parallel [Lindler 1993]. See Figure H.2. No-load temperature differentials of up to 130°C (265°F) can be achieved by stacking modules in this manner. The lowest stages must always have higher heat pumping capacities than the higher stages. Practical multi-stage cascaded modules usually have from 2 to 4 stages, although configurations of up to 6 stages have been built. The analysis of multi-staged cascaded modules can be found in the ITI-Ferrotec catalog [1992] and is included in the accompanying computer program, and will not be discussed further here. Stacked modules are included in the computer code accompanying this report. Stacking does have the effect of decreasing the coefficient of performance with every additional layer, but is only effective when the temperature difference across the module is very large and the COP is quite small, as seen in Figure H.3.

H.4 Power Densities.

A useful quantity for measuring the amount of power produced in a device per unit size is the power density, in the form of power produced per unit area or per unit volume. Power densities achieved to date for a thermoelectric module are in the range of 3 to 6 watts per square centimeter (20 to 40 watts per square inch), which is much higher than in other heating and cooling technologies.

H.5 Heat Pump Construction and Performance.

Construction of a heat pump using these modules is surprisingly easy. (See Figure H.4 for a possible layout.) Direct current power is supplied by means of a stepdown transformer, a rectifier bridge, and a capacitor for smoothing out some of the ripple. The modules can tolerate some ripple in the power supply, so a high degree of filtering is not necessary. Finned heat exchangers are attached to each side of the modules, and the resulting assemblies are then stacked so cold side faces cold side. Half of the layers of air coming through the assembly will then be cooled and the other half heated. Ambient air must be supplied to both sides of each module. Switching from heating to cooling can be done either by mechanically switching from the layers of heated air to the layers of cooled air, or by having just one set of passageways permanently connected to the area to be cooled or heated and simply reversing the direction of the current as needed.

One of the chief design factors in insuring a successful heat pump is the design of the finned heat exchangers. The choice of base thickness and fin height, spacing, and thickness will depend on the air flow provided by the fan. Forced air thermal resistances in the range 0.02°C/W to 0.5°C/W are typical, with a fin base area of 0.04 square meters. The lower the thermal resistance of the heat exchanger, the greater the heat pumping capacity of a given thermoelectric module.

Using representative thermoelectric modules now available, calculations of the refrigeration coefficient of performance were carried out for a cooling load of 20 kilowatts for various thermal resistances of the heat exchanger. The results are shown in Figures H.3 for no heat exchanger thermal resistance, and in Figures H.5 and H.6 including heat exchanger thermal resistance. Figure H.3 was generated by selecting an ambient temperature and varying the current to determine the maximum possible cooling COP. We see from this figure that, with no thermal resistance, for a cool temperature of 22°C (71.6°F) and a hot temperature of 35°C (95°F) a theoretical COP of 2.7 is possible. Basing calculations on Melcor module CP-5-127-06 [Melcor 1993], which has a high area-to-length ratio of the thermoelectric elements (1.255 cm.), approximately 265 modules would be needed to produce 20 kilowatts of cooling. (Cooling/heating power scales linearly with the number of module.)

Figures H.5 and H.6 include the effects of heat exchanger thermal resistance. ITI-Ferrotec modules were used in the calculations. The temperature difference between ambient conditions and the cool side of the heat pump, ΔT, depends on the thermal resistance of the heat exchanger used. Typical values of ΔT are in the 5 to 15°C range. Since for ambient temperatures below 40°C the maximum COP varies steeply with ambient temperature (as seen in Figure H.3) it can be expected that COP will decrease substantially with heat exchanger temperature difference. As an example of the calculations involved, using a heat exchanger with a thermal resistance of 0.02°C/W and a cooling load per heat exchanger of 250 watts, ΔT would be the product of these two numbers, that is, 5°C. Looking at the ΔT=5°C curve in Figure H.5 we see COPs ranging from 1 at 35°C (95°F) to 0.5 at 45°C (113°F). Figure H.6 clearly shows the importance of selecting the correct heat exchanger to maintain reasonable COPs.

H.6 Advantages.

a. Thermoelectric devices are environmentally friendly. No refrigerant is needed. Air may be used for the heat transfer fluid in most room conditioning applications, although in some applications it may be necessary to use liquid heat transfer agents.

b. Changing from cooling to heating requires only a reversal of the direction of current flow or a change in air ducting.

c. Reliabilities higher than 200,000 hours have been obtained in certain applications.

d. Thermoelectric modules are compact in size. The units can be stacked so as to be in parallel thermally and in series electrically to provide larger temperature differences.

e. Capacity modulation is easy, involving adding or subtracting module assemblies as needed.

f. Thermoelectric heating and cooling are electrically and acoustically quiet.

g. Thermoelectric modules can also be used to generate electricity.

h. Phase-change salts have been incorporated in the cold-side heat exchanger as a means of energy storage. Also, solar panels have been used to provide power in some low-energy applications.

H.7 Disadvantages.

a. Thermoelectric efficiencies are lower than CFC-based units.

b. Direct current must be provided, necessitating either rectification of the alternating current supply or a DC generator.

c. Applications to date have been primarily for small sizes, e.g., coolers and home refrigerators [Lackey et al 1958, Mei et al 1993]. Several of the manufacturers contacted said that at this time this technology is not suited to ECUs of the power levels contemplated. Cooling units have been built for tank drivers [Purcupile et al 1968], for commanders of space shuttles, and for individual cooling backpacks. Larger applications by Marvel S. A. in France have included submarines [Stockholm and Schlicklin 1988, 1989, Blankenship et al 1989] and railroad cars [Stockholm et al 1982]. The Marvel units are in the kilowatt range, and one unit for a ship was as high as 100 kilowatts. Jetway Systems of Ogden Utah has used the devices for cooling parked airplanes [Gilliam et al 1992], and is reportedly testing the use of thermoelectric heating/cooling in airport jetways. A prototype13 cubic foot thermoelectric refrigerator prototype was shown at the *Domotechnica* appliance trade show in Germany in February 1995 [Krepchin 1995]. It was built by a consortium consisting of Owens-Corning, Marlow Industries, and Oceaneering Space Systems This refrigerator does not have a freezer compartment, and uses a new highly effective insulation and a phase-change material to level the load. It is said to have an efficiency comparable to the current efficiency leader in Europe, a Bosch propane/butane unit which uses 130 kilowatt hours per year.

d. Most applications to date have been for cases where the thermal load stays nearly constant with time. Cycling (turning the devices on and off) has an adverse effect on reliability. Performance can be sensitive to specific operating conditions.

e. In 1968 reported Z values seldom exceeded 0.005 K^{-1}, with one reported as high as 0.045 K^{-1}. These values are approximately true today, although there are predictions of material breakthroughs within the next five years. A high degree of investigation currently is in progress.

f. Fabrication of modules which have low junction resistance, higher refrigeration capacity, and a higher permissible temperature drop of 100 to 150°F is necessary.

g. Thermal and electrical contact resistance may increase with time. Low contact resistance is essential and contributes appreciably to fabrication costs.

h. Interfaces must provide low electrical and thermal resistivity. In most cases this has required different solutions for every alloy used. Since direct soldering to a semiconductor in many cases is not possible, contacts have to first be plated. In some cases preliminary diffusion of the bonding material into the semiconductor and ion plating has been done.

i. Thermal stresses arising with thermal cycling have in the past caused trouble in maintaining good thermal and electrical contact. Since compression loading has been found to be the simplest in holding elements together, spring loading has been utilized to maintain good contact. Plastic deformation at elevated temperatures and creep rate under constant stress have helped solve the contact problem for polycrystalline ingots of intermetallic compounds.

H.8 Tradeoffs, Hazards, and Risks Analysis.

Since thermoelectric modules have no refrigerant and in the ECU application would use air as the heat transfer fluid, they are environmentally friendly and have no toxic, combustion, or explosive risk. No heat energy need be applied and these devices ordinarily operate within 10°C (18°F) of the ambient and room temperatures, therefore there is only a minimal thermal signature of the device. Mechanical complexity of these heat pumps is low, the acoustic signature is minimal, and there are no moving parts to wear out or which need servicing. The units should be easy to attach to conventional air ducting and fans.

The electrical knowledge needed to repair these devices in the field is relatively basic. In the case of failure of an individual module, that module can easily be removed from the circuit. The usual design would have a number of modules assembled in a panel, so that should a number of modules on the panel fail, the entire panel could easily be replaced. The control system for a thermoelectric heat pump would also be relatively simple. Reliability and storage of thermoelectric modules over long periods of time and in harsh environments is not well proven, but reports in the literature appear favorable. In any case, testing of stored units should be relatively easy, requiring only a standard alternating current source of power.

The present state of development of thermoelectric technology is relatively advanced, with a number of companies in the US and elsewhere actively engaged in research and development, and with a number of small modules (less than 400 watts cooling) available "off the shelf". However, most of the applications have been in low power applications. The present cost of these modules is not price competitive with conventional vapor-compression heat pumps. This may change with the elimination of CFCs and with improved material efficiencies.

H.9 Manufacturers/Suppliers.

Electric & Gas Technology, Inc., 13636 Neutron Rd., Dallas TX 75244-4410, 214-934-8797, fax 214-991-3255.

ITI Ferrotec, 131 Stedman St., Chelmsford MA 01824, 508-452-0212.

Jetway Systems, 3100 S. Pennsylvania Ave., Ogden Utah 84401-3328, 801-627-6600, fax 801-6299-3288.

Marlow Industries, 10451 Vista Park Rd., Dallas TX 75238-1645, 214-340-4900.

Marvel S. A., 11 rue Joachim du Bellay, Marsinval, 78540 Vernouillet, France, 33.1.39.71.69.14.

Melcor (Materials Electronic Products Corp.), 1040 Spruce St., Trenton NJ 08648-1398, 609-393-4178.

Midwest Research Institute, 425 Volker Blvd, Kansas City MO 64110-2299, 816-753-7600.

TECA (Thermoelectric Cooling America Corp.), 4048 W. Schubert St., Chicago IL 60639, 312-342-4900.

TE Technology Inc., 10345-T E. Cherry Bend Rd., Traverse City MI 49684, 616-929-3966.

Tellurex Corp., 1248 Hastings St., Traverse City MI 49684, 616-947-0110.

Thermacore, 780 Eden Rd., Lancaster PA 17601-4271, 717-569-6551.

Thermo Electric Cooling America Corp., 4048 W. Schubert St., Chicago IL 60639, 312-342-4900.

H.10 References.

Angrist, S. W., *Direct Energy Conversion*, 3rd ed., Allyn & Bacon, Boston, 1976.

Blankenship, W. P., C. M. Rose, & P. P. Zemanick, "Application of thermoelectric technology to naval submarine cooling", Proc. 8th Intern. Conf. on Thermoelectric Energy Conversion, July 10-13, Nance France, pp. 224-231, 1989.

Bridgeman, P. W., *The Thermodynamics of Electrical Phenomena in Metals*, MacMillan, New York, 1954.

Cadoff, I. B., & E. Miller, *Thermoelectric Materials and Devices*, Reinhold, New York, 1960.

Chang, S. S. L., *Energy Conversion* (especially chapter 3, Thermoelectric Engines), Prentice-Hall, Englewood Cliffs N. J., 1963.

Egli, P. H., *Thermoelectricity*, John Wiley, New York, 1960.

Eichhorn, R. L., "A review of thermoelectric refrigeration", Proc. IEEE vol. **51**, no. 5, pp. 721-725, 1963.

Gwilliam, S. B., B. Entezam, S. C. Tatton, "Thermoelectric air conditioning using evaporative cooling of waste heat air for parked aircraft", 11th Internat. Conference on Thermoelectrics, pp. 164-174, 1992.

ITI-Ferrotec, *Thermoelectric Products Catalog and Technical Reference Manual #100*, Chelmsford MA, 1992.

Kaye, J., & A. J. Welsh, eds., *Direct Conversion of Heat to Electricity*, especially chapter 21, "Quantitative design of a thermoelectric cooler", John Wiley, New York, 1960.

Krepchin, I. ed., "New Refrigerator Technologies - Thermoelectric cooler has no moving parts", Demand-Side Technology Report vol. 3, no. 9, September 1995.

Lackey, R. S., E. V. Somers, & J. D. Mess, "Application of thermoelectric cooling and heating to novel appliances', Refrig. Engring. vol. **66** no. 12, 1958.

Lindler, K. W., "Improving the performance of thermoelectric heat pumps by use of multi-stage cascades", 28th Intersoc. Energy Convers. Engring Conf Proc, 1993.

MacDonald, D. K., *Thermoelectricity*, John Wiley, New York, 1962.

Mei, V. C., F. C. Chen, B. Mathiprakasam, & P. Heenan, "Study of solar-assisted thermoelectric technology for automobile air conditioning", Trans. ASME vol. **115**, pp. 200-

205, 1993.

Melcor, *Frigichip Thermoelectric Cooling Devices Catalog,* Materials Electronic Product Corp., Trenton NJ 1993.
Penrod, E. B., "A theoretical analysis of a Peltier refrigerator", ASME paper 59-A-266, December, 1959.

Purcupile, J. C., R. E. Stillwagon, & R. E. Franseen, "Development of a two-ton thermoelectric environmental control unit for the U. S. Army", ASHRAE Trans. vol. **74**, part **II**, 1968.

Rosi, F. D., D. Abels, & R. V. Jensen, "Materials for thermoelectric refrigeration", J. Phys. & Chem. Solids vol. **10**, pp. 191-200, 1959.

Soo, S. L., *Direct Energy Conversion*, especially chapter 5, Thermoelectric Systems, Prentice-Hall, Englewood Cliffs N. J., 1968.

Stockholm, J. G., L. Pujol-Soulet, & P. Sternat, "Prototype thermoelectric air conditioning of a passenger railway coach", Proc. 4th Intern. Conf. on Thermoelectric Energy Conversion, U Tx @ Arlington, Mar. 10-12, pp.135-141, 1982.

Stockholm, J. G. & P. M. Schlicklin, "Thermoelectric cooling for naval applications", Proc. 7th Intern. Conf. on Thermoelectric Energy Conversion, U TX at Arlington, Mar. 16-18, pp. 79-84, 1988.

Stockholm, J. G. & P. M. Schlicklin, "Naval thermoelectrics", Proc. 8th Intern. Conf. on Thermoelectric Energy Conversion, Nance France, Jul. 10-13, pp. 235-246, 1989.

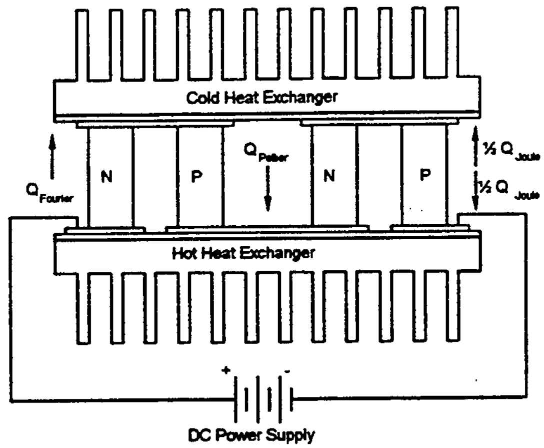

Figure H.1. Thermoelectric element heat flow.

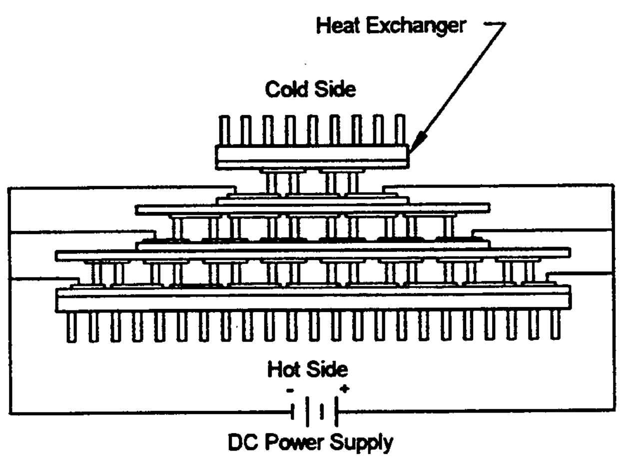

Figure H.2. Cascaded thermoelectric module.

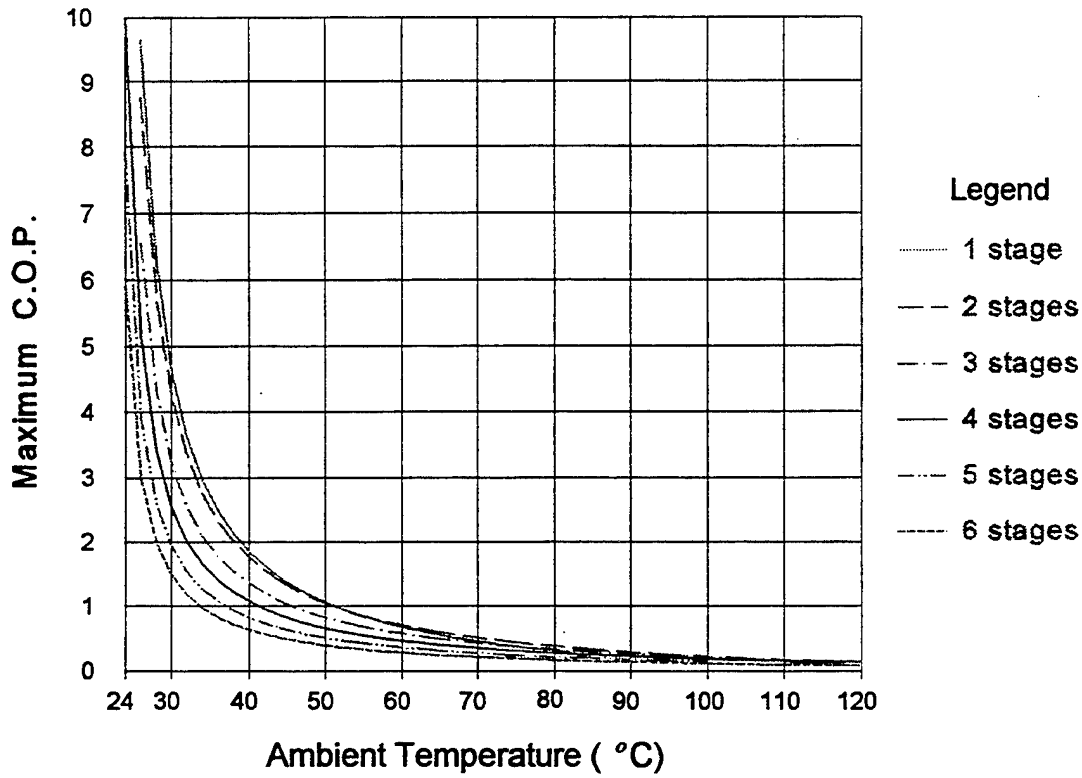

Figure H.3. Maximum cooling COP as a function of ambient temperature for multi-staged modules. Cool temperature is 22°C. Single-stage module is Melcor CP-5-127-06 with a geometric factor of 1.255.

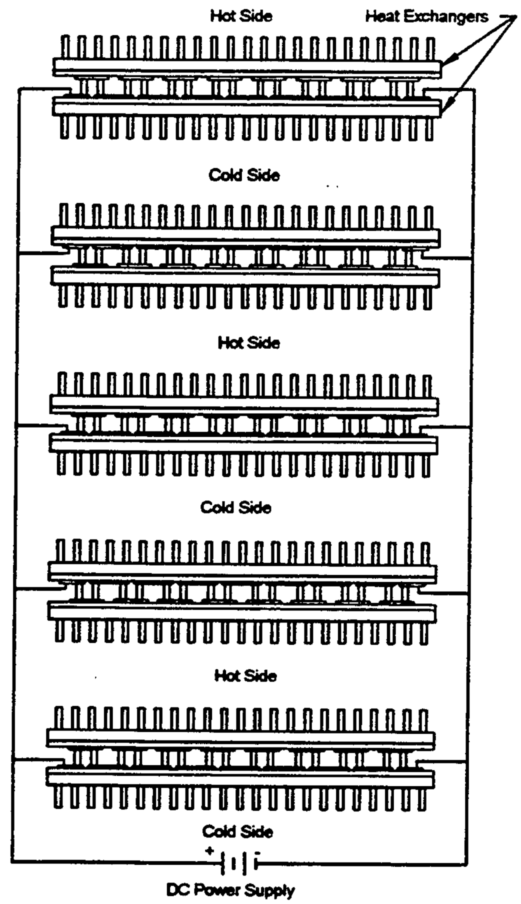

Figure H.4. Possible thermoelectric heat pump layout.

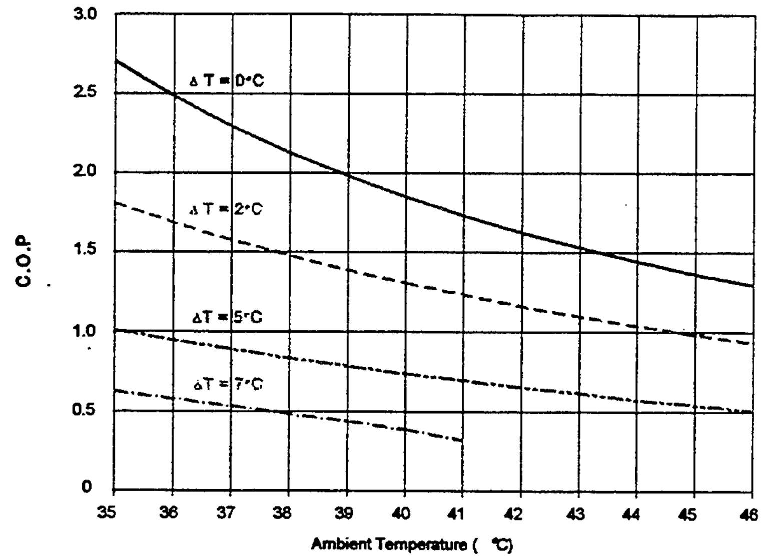

Figure H.5. Refrigeration COP as a function of ambient temperature. Curves of constant heat exchanger ΔT.

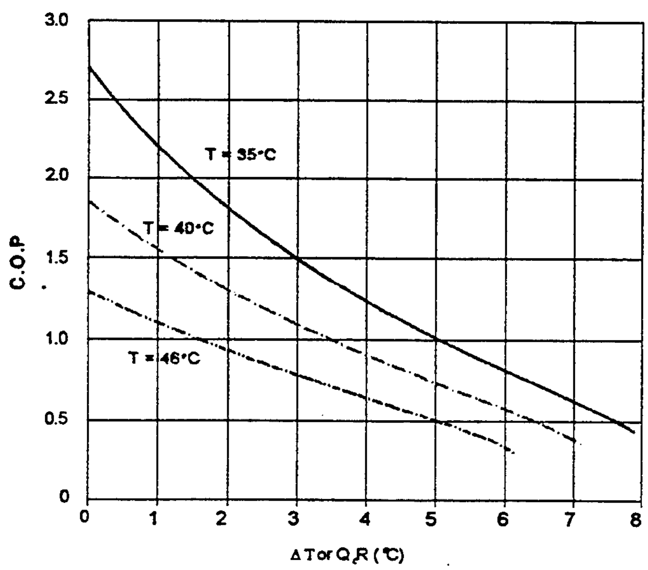

Figure H.6. Refrigeration COP as a function of heat exchanger ΔT. Curves of constant ambient temperature.

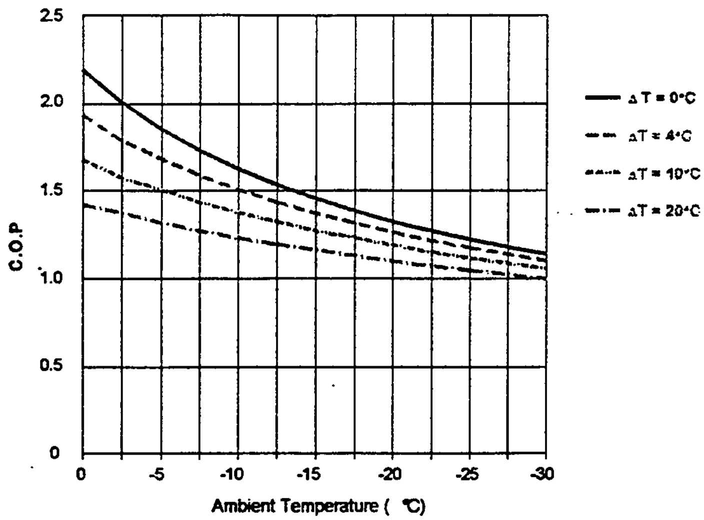

Figure H.7. Heating COP as a function of ambient temperature. Curves of constant ΔT.

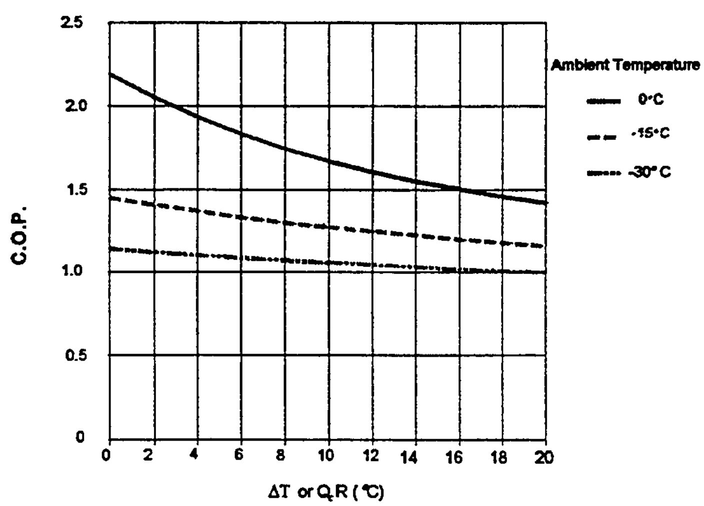

Figure H.8. Heating COP as a function of ΔT. Curves of constant ambient temperature.

I. VAPOR-COMPRESSION HEAT PUMPS.

I.1 General Description.

The vapor compression cycle is the most common cycle used in heating and cooling applications. The high efficiency and simple embodiment of this cycle make it very attractive. A vapor compression heat pump consists of a mechanical compressor, condenser, expansion valve or capillary tube, and evaporator. The compressor power source is commonly a standard 110 or 220 VAC power source.

Refrigerants used are typically a CFC (R-12), HFC (R-134a), or ammonia. Both CFCs and HFCs have high efficiencies and are non-toxic, but CFCs have been deemed harmful to the environment, depleting the ozone layer. Any of the listed refrigerants allow the devices to operate over the temperature ranges associated with heat pumps and refrigerators. CFCs have the highest efficiencies as well as a high affinity for oil, which aids in the lubrication of the mechanical components.

The cycle is relatively simple and can be evaluated by basic thermodynamic equations. The compressor mechanical efficiency and enthalpy changes are related according to

$$\eta = (h_{out,s} - h_{in})/(h_{out} - h_{in}) \qquad (I.1)$$

The rate of heat loss or gain in any portion of the cycle can be determined from

$$Q = \dot{m}\Delta h. \qquad (I.2)$$

The throttling process across the expansion valve/capillary tube is such that the refrigerant enthalpy remains constant, or

$$h_{in} = h_{out}. \qquad (I.3)$$

The refrigeration COP is given by

$$COP_{refrigeration} = Q_{evaporator}/W_{compressor}. \qquad (I.4)$$

With the use of the expression for efficiency given in equation (I.1) above this COP does account for the losses involved in converting electrical energy into compressor operation.

The vapor compression cycle and a schematic diagram are shown in Figures I.1 and I.2. The refrigerant vapor enters the compressor which mechanically compresses the gas, thereby increasing its pressure. This pressure increase also increases the gas temperature. The vapor proceeds to the condenser where heat is rejected to the ambient temperature reservoir, condensing the refrigerant to a liquid. The liquid is then passed through an expansion valve, decreasing its temperature. The liquid then enters the evaporator where heat is absorbed from the room temperature reservoir, allowing the refrigerant to change phase back to a vapor. This vapor re-enters the compressor repeating the cycle.

Under this contract this cycle has been programmed for two refrigerants, ammonia and R-134a. Results quoted for R-11, R-12, R-22, and R-500 were obtained from use of a commercial code.

I.2 Advantages.

a) The COPs for all the refrigerants is relatively high.

b) The apparatus is simple and compact.

c) The technology has been highly characterized.

d) Valves switch the operating mode from heating to cooling.

e) The device requires only a conventional power supply.

I.3 Disadvantages.

a) Some of the refrigerants (CFCs) are environmentally unfriendly.

b) Ammonia, while widely used today in commercial refrigeration, is irritating in small doses and toxic in larger amounts. It does have a concentration range in which it is flammable. In spite of these problems, serious injuries involved with ammonia refrigeration are rare.

c) R-134a does not contain chlorine and hence does not destroy ozone. However it does not mix well with standard lubricants, may contribute to global warming, and has shown itself to require substantial technological design changes in HVAC systems. Its long term characteristics are still in question.

I.4. Tradeoffs, Hazards, and Risks Analysis.

The vapor-compression cycle is highly efficient and years of development have brought it to an inexpensive and reliable system. However, many if not most of the commonly used refrigerants cause problems. Besides the ozone depleting CFCs, ammonia is a toxic gas and exposure can be harmful or even lethal to personnel should leaks occur.

The refrigerant COPs for six of these refrigerants were computed using a commercial program for the CFCs and HFCs, and new programs developed under the contract for ammonia and R-134a. A comparison of six refrigerants under equal operating conditions (room temperature of 22°C and 80% compressor efficiency) is seen in Figure I.3. The evaporator and condenser are assumed to be 100% efficient in these computations. It is seen that R-11 is the most efficient, followed by ammonia, R-12, R-500, R-22, and R-134a, in that order.

To see the effect of condenser and evaporator efficiency on the cycle operation, Figures I.4 through I.7 show the COP as a function of the ambient temperature and ΔT, the difference between the reject temperature and the condenser or evaporator. In all cases the refrigerant is R11, the room temperature is 22°C, and the compressor efficiency is 80%. In all cases a ΔT of zero (a perfect heat exchanger) shows the highest COP. An efficient heat exchanger would have a low thermal resistance (defined as the ratio of ΔT to the heat transferred). Figures I.4 and I.5 show the refrigeration COP, and Figures I.6 and I.7 show the heating COP. The lowering of COP with increasing ΔT shows the importance of correct heat exchanger design to these heat pumps. Similar results are obtained for other heat pump technologies.

I.5 Common Refrigerants used in Vapor-Compression.

R-11. Trichloromonofluoromethane (CCI3F, a CFC). Stable, nontoxic, nonflammable, used typically with centrifugal compressors at high temperatures, low pressures. Applications include: chillers, automotive refrigeration, residential cooling, foam blowing of polyurethanes.

R-12. Dichlorodifluoromethane (CCl_2F_2, a CFC). Colorless, odorless, nontoxic, noncorrosive, nonirritating, nonflammable. Has a low latent heat which is advantageous in smaller refrigeration systems at medium temperatures. Uses reciprocating, centrifugal, or rotary compressors. Applications include: chillers, automotive air conditioning, residential cooling.

R-22. Monochlorodifluoromethane ($CHClF_2$, an HCFC). Stable, nontoxic, noncorrosive, nonirritating, nonflammable. Used for low temperature applications.

R-134a. Tetrafluoroethane (CF_3CH_2F, an HFC). Contains no chlorine, therefore causes no ozone depletion. Not compatible with present lubricants.

R-500. An azeotropic mixture of R-12, (26.27%) and R-152a (CH_3CHF_2). Used in larger industrial and commercial applications having reciprocating compressors. Has a greater refrigerating capacity than R-12 alone.

R-717. Ammonia (NH_3). Colorless, somewhat flammable, with proper proportions of air can form an explosive mixture. Has a pronounced and distinguishable odor. Used in industrial systems at low temperatures.

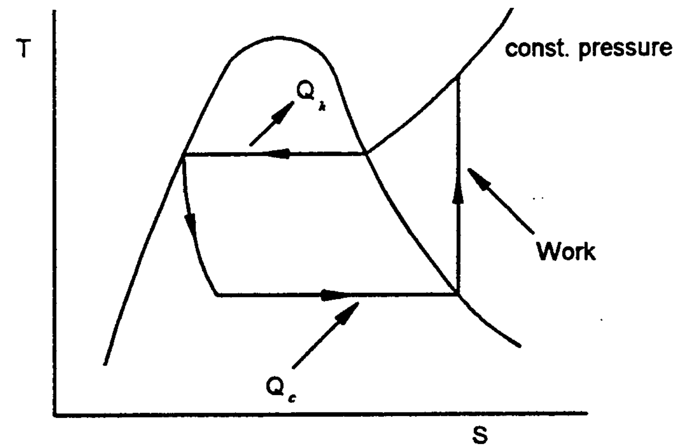

Figure I.1. Vapor-compression cycle temperature-entropy diagram.

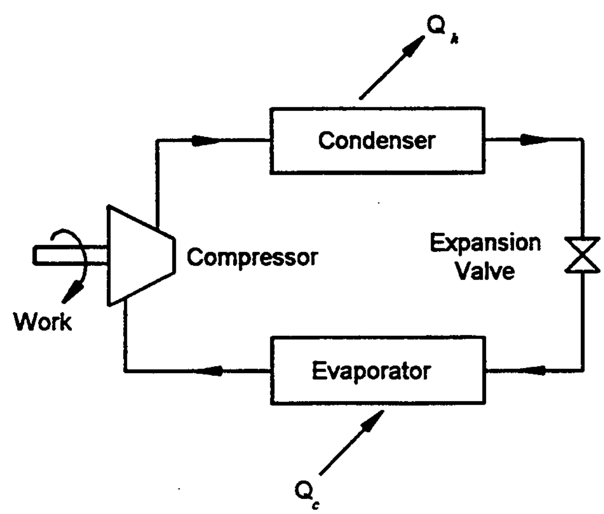

Figure I.2. Vapor-compression cycle schematic.

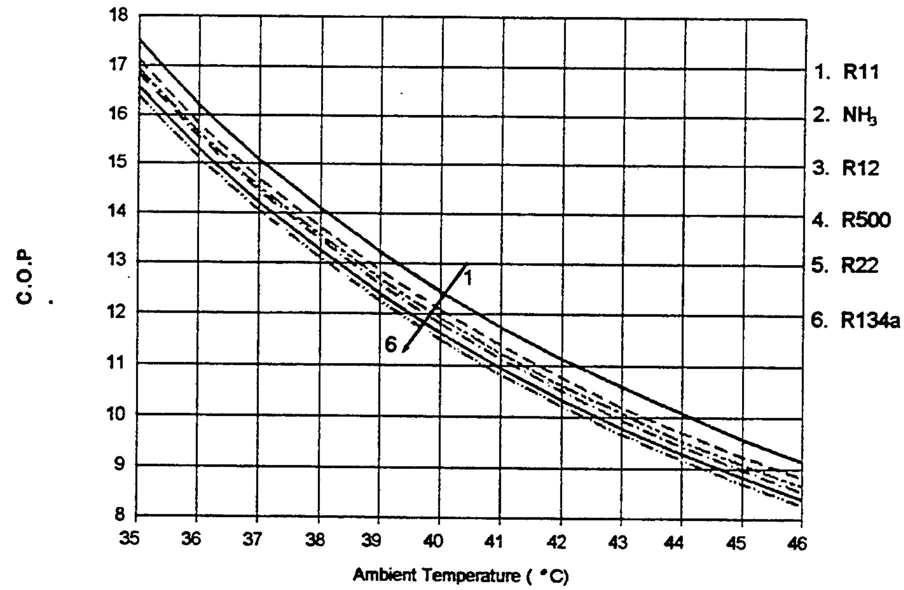

Figure I.3. Vapor-compression cycle refrigeration COP versus ambient temperature for six refrigerants.

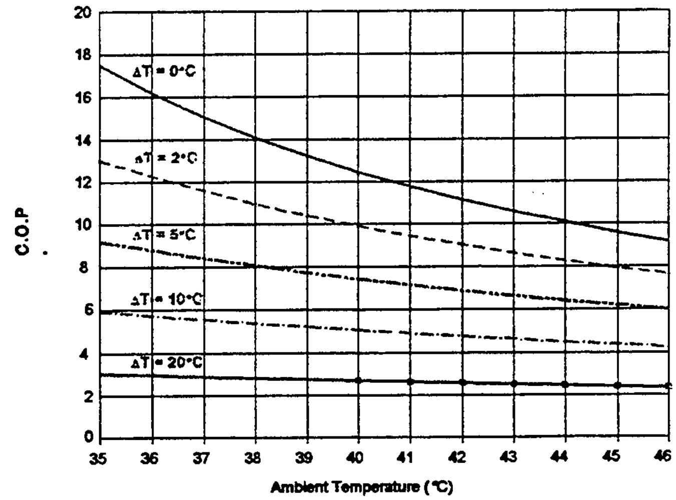

Figure I.4. Refrigeration COP as a function of ambient temperature for R-11. Curves of constant ΔT.

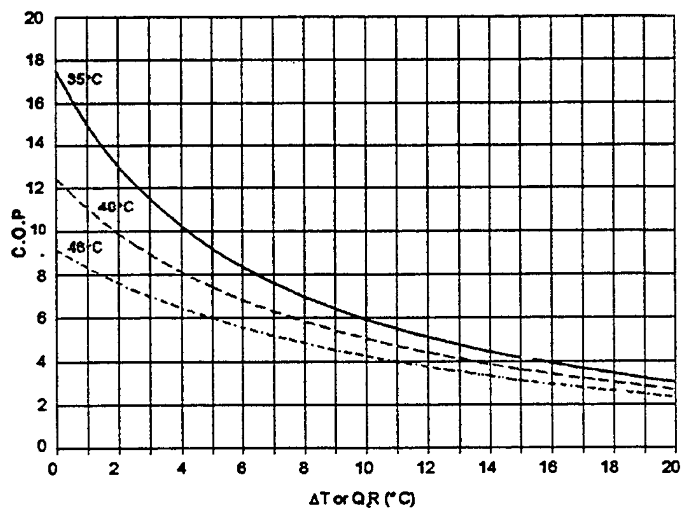

Figure I.5. Refrigeration COP as a function of ΔT for R-11. Curves of constant ambient temperature.

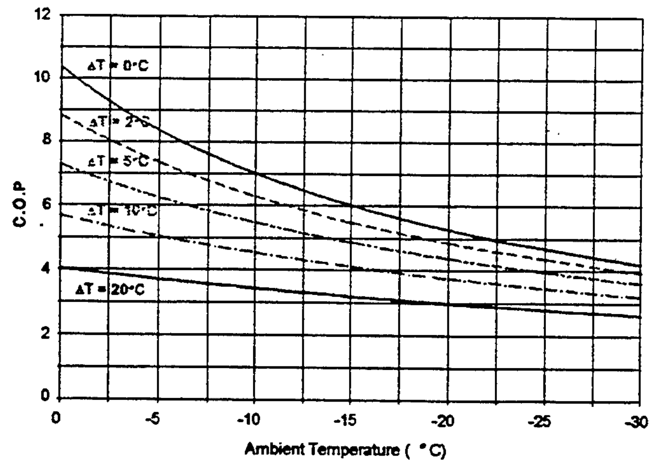

Figure I.6. Heating COP as a function of ambient temperature for R-11. Curves of constant ΔT.

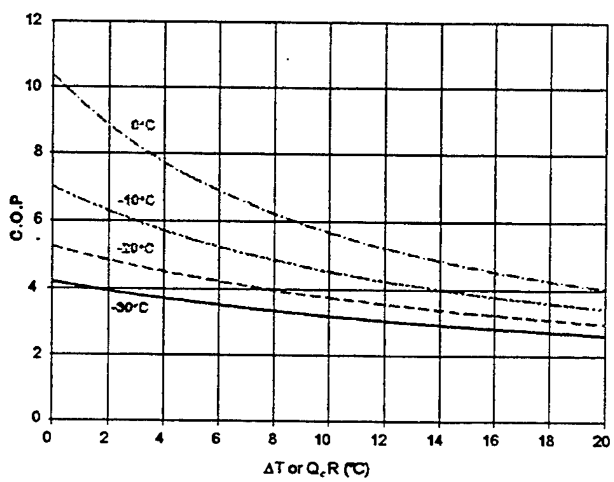

Figure I.7. Heating COP as a function of ΔT for R-11. Curves of constant ambient temperature.

J. Vortex Tube Heat Pumps.

J.1 General Description.

A vortex tube (also called a Ranque tube or a Hilsch tube, after its original discoverers [Ranque 1933, Hilsch 1947]) consists of a round tube with an inlet about one-quarter of its length from one end (the "cold end"). See Figure J.1. High pressure compressed air typically at 100 psig is introduced into the main body of the tube tangent to the tube axis, thereby introducing a high degree of swirl velocity to the entering air. Air velocity is at or near sonic conditions after it enters the tube. The exact behavior of the air inside the tube is complex and requires detailed analysis [Kurosaka 1982]. Some of the details of the physical mechanisms are still unclear.

A restriction in the diameter at the cold end of the entrance chamber causes most of the air to originally move toward the longer part of the tube (the "hot end"). At the hot end the air encounters a control needle valve. At the location where the air first encounters the control valve, the air pressure is less than the pressure of the exiting air, but still higher than atmospheric pressure.

Setting the control valve at the hot end exit determines what percentage of the air exits at the hot end, and how much air is returned to the cold end. The fraction of flow exiting the cold end is termed the "cold fraction" (CF). Air returned to the cold end of the tube is forced to the center of the tube, where it traverses the hot tube, the entrance chamber, and the cold tube on its way to the cold exit.

The two concentric air streams in the hot tube have the same swirl velocity. In the hot region of the vortex tube, heat flows from the inner core of the flow to the outer portion of the flow. All of the flow in the inner core of the vortex tube exits at the cold end of the vortex tube. No net heat is generated inside the vortex tube. Heat gained by the air exiting at the hot end was lost by the air that exits at the cold end.

The following table and formulae give the characteristics of the vortex tubes produced by the Vortec Corporation, the principal manufacturer of these devices. Air consumption for their tubes ranges from two standard cubic feet of air per minute (SCFM) to 100 SCFM at an inlet pressure of 100 pounds per square inch gauge (psig). Heat capacities range from 100 Btu per hour (Btu/hr) to 6,000 Btu/hr. Their physical lengths range from about 6-1/2 inches to 13-1/2 inches. For a 100 psig system, the Vortec Corporation [Vortec1990] suggests that it takes approximately one horsepower to compress 4 SCFM of air.

The bold numbers in Table J.1 give the temperature drop (°F) of the air leaving the cold end of the vortex tube, and the italic numbers give the temperature rise (°F) of the air leaving the hot end. Both temperature rise and drop are functions of the inlet pressure and the cold fraction. The cold fraction is determined by adjustment of the control valve, and in any setup would be most easily found by measuring the inlet, hot, and cold temperatures and then determining the cold fraction from

$$CF = (T_{hot} - T_{inlet} + 4)/(T_{hot} - T_{cold}) \quad (J.1)$$

with the temperatures in °F.

Table J.1. Vortex tube performance data (from Vortec Corporation catalog)

Inlet pressure psig		Cold fraction %						
		20%	30%	40%	50%	60%	70%	80%
20 psig	cold	**61.5**	**59.5**	**55.5**	**50.5**	**43.5**	**36.0**	**27.5**
	hot	*14.5*	*24.5*	*36.0*	*49.5*	*64.0*	*82.5*	*107*
40 psig	cold	**88.0**	**85.0**	**80.0**	**73.0**	**62.5**	**51.5**	**38.0**
	hot	*20.5*	*35.0*	*51.5*	*71.0*	*91.5*	*117*	*147*
60 psig	cold	**104**	**100**	**93.0**	**84.0**	**73.0**	**59.5**	**44.5**
	hot	*23.5*	*40.0*	*58.5*	*80.0*	*104*	*132*	*168*
80 psig	cold	**115**	**110**	**102**	**92.0**	**80.0**	**65.5**	**49.0**
	hot	*25.0*	*43.0*	*63.0*	*86.0*	*113*	*143*	*181*
100 psig	cold	**123**	**118**	**110**	**99.0**	**86.0**	**70.5**	**53.0**
	hot	*26.0*	*45.0*	*66.5*	*91.0*	*119*	*151*	*192*
120 psig	cold	**129**	**124**	**116**	**104**	**90.5**	**74.0**	**55.0**
	hot	*26.0*	*46.0*	*69.0*	*94.0*	*123*	*156*	*195*
140 psig	cold	**135**	**129**	**121**	**109**	**94.0**	**76.0**	**56.5**
	hot	*25.5*	*46.0*	*70.5*	*96.0*	*124*	*156*	*193*

The numbers in Table J.1 giving the temperature rise and drop are based on an inlet temperature of 70°F. They can be corrected for other inlet temperatures by multiplying by the temperature correction factor (tcf) determined from

$$tcf = (T_{inlet} + 460)/530, \quad (J.2)$$

where T_{inlet} is the actual inlet temperature in degrees Fahrenheit.

The total flow through any of these vortex tubes depends on the inlet pressure, which can be controlled by a valve. Total flow is given as the tube flow rating at 100 psig times the absolute inlet pressure (gauge pressure plus 15) divided by 115, or

total flow in (SCFM) =

$$(\text{rated flow @ 100 psig})\cdot(\text{inlet psig}+15)/115. \qquad (J.3)$$

The temperature rise (T_{hot} - T_{inlet}) at the hot end and the temperature drop (T_{inlet} - T_{cold}) at the cold end are given in Table J.1. From these the heating and cooling power can be found from

$$Btu/hr_{heating} = 1.0746\cdot CFM_{total}\cdot(1 - CF)\cdot(T_{hot} - T_{inlet}), \qquad (J.4)$$

$$Btu/hr_{cooling} = 1.0746\cdot CFM_{total}\cdot CF\cdot(T_{inlet} - T_{cold}), \qquad (J.5)$$

where CF is the cold fraction expressed as a decimal number between 0 and 1.

Maximum heating and cooling both take place when the cold factor is about 60%. From consideration of the temperature drop as a function of inlet pressure it is found that the maximum temperature drop occurs at the lowest cold factors. From the temperature rise versus inlet pressure figure it is seen that maximum temperature rise occurs at the highest cold factors.

From the above equations it is possible to give an estimate of the compressor size needed for vortex tube cooling and heating. Using a cold factor of 0.6 and a compressor providing 100 psig, from Table J.1 it is seen that the hot temperature difference is 119°F and the cold temperature difference is 86°F. Using an estimate of 1 horsepower to compress 4 CFM and equation 4, to provide 10,000 Btu per hour of heating would require 195.5 CFM of air and 48 compressor horsepower. For 1 ton of cooling, equation 5 predicts a requirement of 216.4 CFM and 54.1 horsepower. Since the requirements for the ECU are 55 thousand Btu per hour for heating and 5 tons of cooling, compressor sizes of at least 270 horsepower would be required.

J.2 Advantages.

a. The vortex tube is mechanically an extremely simple device, requiring little maintenance if a properly filtered air supply is provided.

b. Their output can be controlled by simply adjusting the inlet valve either manually or automatically.

c. They can be cascaded in parallel to increase the degree of heating and cooling.

d. They can be arranged so that the hot ends of the vortex tubes all exit to one duct (the "hot duct"), and the cold ends all exit to another duct (the "cold duct").

e. Changing from heating to cooling requires only switching room air from the hot duct to the cold duct. The hot duct is then dumped to the exterior. Changing from cooling to heating reverses this. Once the control needle valve is set for an individual vortex tube, little or no further adjustment is necessary.

f. No refrigerants are needed. Leakage and storage are not problems. No external heat exchangers are needed since the heating/cooling system is an open one.

g. No recirculation of room air occurs, resulting in minimum interior air pollution.

J.3 Disadvantages.

a. The compressors needed for the compressed air are bulky and generally of low efficiency. To produce sufficient cooling to meet ECU requirements would require compressors in the 270 horsepower range.

b. Noise is emitted from the vortex tubes. This can be eliminated by proper lining of air ducts or sound absorbing baffles.

c. Entering air must be clean and free of moisture. Dirt will degrade the performance of a vortex tube. It is customarily removed by passing the compressed air through five micron filters, which must be serviced. Air compressors typically introduce oil droplets into the air line which must be removed by traps or similar devices. Moisture in the air supply will result in frost or ice build-up at the cold end of the tube, degrading its performance. Traps or dryers are customarily used to remove moisture.

d. No recirculation of room air results in lower efficiency of the system because of loss of heated or cooled interior air.

e. Because of the large supply of compressed air needed, these devices have not been used to date for cooling volumes larger than protective hoods or clothing. Practical room cooling depends on the development of more efficient, lighter weight compressors.

J.4 Tradeoffs, Hazards, and Risks Analysis.

There is no inherent risk to either personnel or the environment in the vortex tubes per se, as they use air as the refrigerant. Since the compressors operate at relatively high pressures (100 psig), there is some risk in the compressors and connecting lines. Because the size and capacity of the compressors needed for ECU cooling and heating is large, this technology is not suited to Air Force needs.

J.5 Manufacturers.

Exair Corp., 1250 Century Circle North, Cincinnati OH 45246, 513-671-3322.

Vortec Corporation, 10125 Carver Rd., Cincinnati OH, 45242-9976, 800-441-7475.

J.6 References.

Hilsch, R., Rev. Sci. Instrum. vol. **18**, pp. 108-113, 1947.

Kurosaka, M., "Acoustic Streaming in swirling flow and the Ranque-Hilsch (vortex-tube) effect", J. Fluid Mech. vol. **124**, pp. 139-172, 1982.

Ranque, G. J., J. Phys. Radium vol. **4**, pp. 1128-1158, 1933.

Vortec Corporation, *Short Course on the Vortex Tube*, 1990.

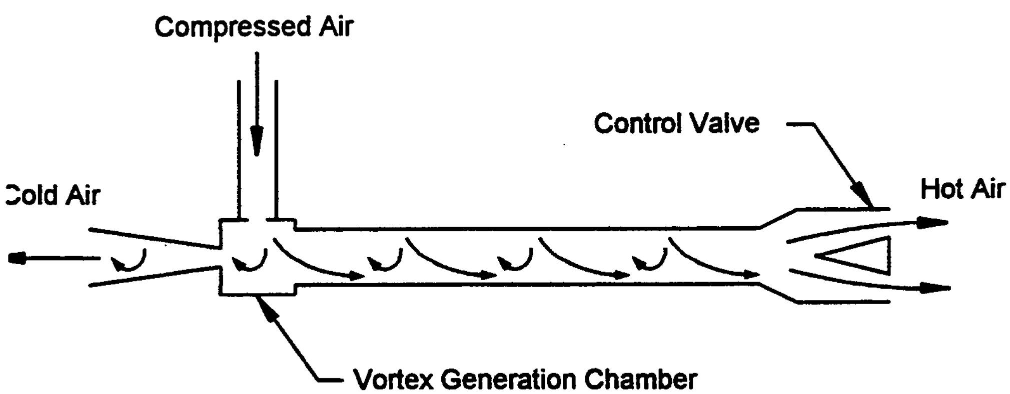

Figure J.1. Vortex tube schematic.

K. Fortran Programs.

This is a catalog of the FORTRAN programs developed under this contract. Sample outputs are given for suggested data in the individual programs. Data may be input to the program in both SI and English units. Results are presented in the units selected. In all cases the results are displayed both on the monitor and stored in a data file on a drive chosen by the operator. Output files in some cases give more data than is shown in this catalog.

K.1 Acoustic Heat Pump Program - ACOUSTIC.FOR

This acoustic heat pump model allows the operator to select between six gases. The data to be input is the following:

Drive to be used for output files

Choose between:

1. Compute the driver frequency for a suitable combination of tube length and heat pumping capacity.
2. A typical heat pump consists of a driving piston, a larger diameter tube, a smaller diameter tube, and a resonator chamber. Using the length and frequency from option 1, design the lengths of the second tube and size the volume of the resonator.

Under option 1:

- Units
- Mean pressure and temperature
- Gas
- Plate thermal conductivity, mass density, specific heat, spacing, thickness, length
- Tube diameter
- Temperature at hot and cold ends
- Peak acoustic pressure
- Frequency range

Under option 2:

- Units
- Mean pressure and temperature
- Gas
- Peak acoustic pressure
- Diameter of tube nearest driver
- Diameter of second tube
- Length of second tube

The program then calculates the size of the resonator and acoustic energy losses.

Sample output for ACOUSTIC.FOR - SI units

Option 1

Gas is HELIUM.	
Acoustic speed is	941.8 [m/s]
Mass density is	1.880 [kg/m^3]

Ratio of specific heats is 1.667
Prandtl number is .674
Thermal conductivity is .138 [W/m-K]
Specific heat C_p is 5193.00 [kJ/kg-K]
Thermal conductivity of the plate is .16 [W/m-K]
Mass density of the plate material is 1400.00 [kg/m^3]
Specific heat of the plate material is 1.10 [kJ/kg-K]
Plate spacing is .380 [mm]
Plate thickness is .160 [mm]
Plate length is 80.000 [mm]
Tube diameter is 38.000 [mm]
Hot end temperature is 28.0 [deg C]
Cold end temperature is -62.0 [deg C]
Temperature difference along the plate is 90.00 [deg C]
Acoustic pressure amplitude is 30.000 [kPa]
Carnot COP = 2.3461

Frequency	circular frequency	wave length	quarter wave length
[hz]	[radians/second]	[m]	[m]
500.	3141.593	1.884	.471

Perimeter	deltav	deltak	deltap	plate heat cap ratio
[m]	[mm]	[mm]	[mm]	
4.200	.078	.095	.257	2.337

Option 2

Gas is HELIUM.
Acoustic speed is 941.8 [m/s]
Mass density is 1.880 [kg/m^3]
Ratio of specific heats is 1.667
Prandtl number is .674
Thermal conductivity is .138 [W/m-K]
Specific heat Cp is 5193.00 [kJ/kg-K]

Frequency	wave length
[hz]	[mm]
500.	1883.51

Circular frequency	quarter wave length	wave number
[radians/second]	[mm]	[1/m]
3141.59	470.88	3.34

Diameter of the tube nearest the driver is 38.00 [mm]
Diameter of the second tube is 19.00 [mm]
Volume of the resonator is 52.93 [cm^3]
First tube length = 470.88 [mm]
Second tube length = 470.00 [mm]
Total tube length = 940.88 [mm]
Frequency = 500.0 [hz]
Loss normalizing factor = .59056
Losses of energy in acoustic heat pump [W]

First tube	second tube	Resonator	Total
1.68146	.84073	.01322	2.53541

Sample output for ACOUSTIC.FOR - English units

Option 1

Gas is HELIUM.
Acoustic speed is 3089.7 [ft/s]
Mass density is .117 [lbm/ft^3]
Ratio of specific heats is 1.667

Prandtl number is	.674
Thermal conductivity is	.080 [Btu/hr-ft-R]
Specific heat Cp is	1240.33 [Btu/lbm-R]
Thermal conductivity of the plate is	.09 [Btu/hr-ft-R]
Mass density of the plate material is	87.40 [lbm/ft^3]
Specific heat of the plate material is	.26 [Btu/lbm-R]
Plate spacing is	.015 [in]
Plate thickness is	.006 [in]
Plate length is	3.150 [in]
Tube diameter is	1.500 [in]
Hot end temperature is	82.4 [deg F]
Cold end temperature is	-80.0 [deg F]
Temperature difference along the plate is	162.40 [deg F]
Acoustic pressure amplitude is	4.350 [psia]
Carnot COP =	2.3379

Frequency [hz]	circular frequency [radians/second]	wave length [ft]	quarter wave length [ft]
300.	1884.955	10.299	2.575

Perimeter [ft]	deltav [in]	deltak [in]	deltap [in]	plate heat cap. ratio
13.827	.004	.005	.013	2.335

Option 2

Gas is HELIUM.

Acoustic speed is	3089.7 [ft/s]
Mass density is	.117 [lbm/ft^3]
Ratio of specific heats is	1.667
Prandtl number is	.674
Thermal conductivity is	.080 [Btu/hr-ft-R]
Specific heat Cp is	1240.33 [Btu/lbm-R]

Frequency [hz]	wave length [in]
500.	74.15

Circular frequency [radians/second]	quarter wave length [in]	wave number [1/ft]
3141.59	18.54	10.94

Diameter of the tube nearest the driver is	1.50 [in]
Diameter of the second tube is	.75 [in]
Volume of the resonator is	3.24 [in^3]
First tube length =	18.54 [in]
Second tube length =	18.54 [in]
Total tube length =	37.08 [in]
Frequency =	500.0 [hz]
Loss normalizing factor =	.59204

Losses of energy in acoustic heat pump [Btu/hr]

First tube	second tube	Resonator	Total
5.75173	2.87587	.03130	8.65890

K.2 Complex compound heat pump program - AMMONIA.FOR

This model of a complex compound heat pump allows the operator to set any combination of evaporator and condenser temperatures or pressures. By inputting the desired cooling or heating load and specifying the Nernst properties of the complex compound, the amount of complex compound needed and also the average flow rate of ammonia is determined.

The properties of ammonia used in the program are taken from formulas given in W. C. Reynolds; *Thermodynamic Properties in SI*, Stanford University 1979.

The data to be input is the following:

- Drive to be used for output files
- Desired units
- Options:
  - Given ammonia temperature and volume, find pressure
  - Given ammonia temperature and pressure, find volume
  - Given ammonia pressure and volume, find temperature
  - Vapor-compression cycle
  - Complex compound cycle
- Evaporator temperature or pressure
- Condenser temperature or pressure
- (The following is for the complex compound option)
- Nernst equation parameters
- Desorption and adsorption temperatures
- Size based on cooling or heating load
- Hour-averaged desorption rate
- Molecular weight of salt
- Unrecoverable energy needed to raise salt and reactor vessel from absorption temperature to desorption temperature

Provided that a reasonable assessment of condenser and evaporator losses and also the irrecoverable heat loss incurred in heating the reactor vessels are included, the coefficients of performance computed are realistic.

Sample output for AMMONIA.FOR - SI units

```
AMMONIA VAPOR-COMPRESSION & COMPLEX COMPOUND
       HEAT PUMP ANALYSIS.

   COMPLEX COMPOUND ABSORPTION CYCLE
Your complex compound is described by the equation
log (p) = a/T + b, where p is the NH3 pressure [kPa]
and the log is to base 10. For your complex compound
a = -2120.77500, b =    8.87117
The temperature is in [deg K].
The desorbing temperature of the complex compound must be
greater than    96.28 [deg C]
The absorbing temperature of the complex compound must
```

be less than 85.68 [deg C]

	EVAPORATOR		COMPLEX COMPOUND		CONDENSER		
Pressure	913.797		913.602	1350.527		1350.477	[kPa]
	INLET	OUTLET	ABSORB	DESORB	INLET	OUTLET	
Temp.	22.00	22.00	85.67	96.28	49.58	35.00	[deg C]
Volume	.0091	.1403	.1825	.1249	.103405	.001702	[m^3/kg]
Enthal.	492.71	1607.18	1777.51	1789.09	1659.25	492.71	[kJ/kg]
Entropy	1.9604	5.7363	6.2608	6.1125	5.7403	1.9534	[kJ/kg-K]
Quality	.0535	sat vap	vapor	vapor	sat vap	liquid	

The molecular weight of your salt is 155.000
The average hourly desorption rate of your salt is
7.00 [moles ammonia/moles salt-hr]
For the thermal load you selected the required
ammonia mass flow rate during sorption is 64.605 [kg/hr]
For the above ammonia mass flow rate you will need
84.001 [kg] of salt in each vessel.
The heat needed to absorb or desorb the complex compound
per kilogram of complex compound is, according to
the Nernst equation, 2.384 [kJ/kg]
Desuperheater heat reject -2.330 [kW]
Condenser heat reject -20.934 [kW]
Capillary/valve heat reject .000 [kW]
Evaporator heat load 20.000 [kW]
Subcooler heat load 3.057 [kW]
Absorb/desorb heat rate 42.792 [kW]
Vessel/salt heat load 9.601 [kW]
Unrecoverable energy added to vessel/salt
= 535000. [J]
COPcool = .361, COPheat = .416
EERcool = 1.231, EERheat = 1.419

Sample output for AMMONIA.FOR - English units

AMMONIA VAPOR-COMPRESSION & COMPLEX COMPOUND
HEAT PUMP ANALYSIS.

COMPLEX COMPOUND ABSORPTION CYCLE
Your complex compound is described by the equation
log (p) = a/T + b, where p is the NH3 pressure [psia]
and the log is to base 10. For your complex compound
a = -3821.81300, b = 11.87486
The temperature is in [deg R].
The desorbing temperature of the complex compound must be
greater than 205.64 [deg F]
The absorbing temperature of the complex compound must
be less than 186.56 [deg F]

	EVAPORATOR		COMPLEX COMPOUND		CONDENSER		
Pressure	132.535		132.368	196.089		195.870	[psia]
	INLET	OUTLET	ABSORB	DESORB	INLET	OUTLET	
Temp.	71.60	71.60	186.50	205.70	121.24	95.00	[deg F]
Volume	.1451	2.2473	2.9284	2.0000	1.656390	.027267	[ft^3/lbm]

Enthal.	211.84	691.01	764.43	769.44	713.39	211.84	[Btu/lbm]
Entropy	.4682	1.3701	1.4958	1.4602	1.3710	.4666	[Btu/lbm-R]
Quality	.0535	sat vap	vapor	vapor	sat vap	liquid	

The molecular weight of your salt is 155.000
The average hourly desorption rate of your salt is
    7.00 [moles ammonia/moles salt-hr]
For the thermal load you selected the required
    ammonia mass flow rate during sorption is 142.429 [lbm/hr]
For the above ammonia mass flow rate you will need
    185.190 [lbm] of salt in each vessel.
The heat needed to absorb or desorb the complex compound
    per kilogram of complex compound is, according to
    the Nernst equation, 1.026 [Btu/lbm]
Desuperheater heat reject -7982.797 [Btu/hr]
Condenser heat reject -71434.670 [Btu/hr]
Capillary/valve heat reject .000 [Btu/hr]
Evaporator heat load 68246.000 [Btu/hr]
Subcooler heat load 10457.440 [Btu/hr]
Absorb/desorb heat rate 146187.100 [Btu/hr]
Vessel/salt heat load 32756.160 [Btu/hr]
Unrecoverable energy added to vessel/salt
    = 507. [Btu]
COPcool = .360, COPheat = .416
EERcool = 1.229, EERheat = 1.418

K.3 Stirling Cycle Heat Pump Programs.

Stirling cycle heat pumps consist of fluid-containing working volumes separated by pistons or displacers (a piston with equal pressure on both sides), heat exchangers, and a regenerator. The regenerator is a heat exchanger which acts as a thermal storage device, absorbing thermal energy for release later during the cycle.

These definitions and conventions hold in the following programs:

All gas in a given space is considered to be at that space's temperature (isothermal assumption). The gas in a particular chamber is also assumed to all be at the same pressure (uniform pressure). Except for program STIRLNG6, the pistons-displacers are assumed to move in a sinusoidal manner.

The void volume (also called the dead space) is that portion of the total volume which is not swept by the piston or displacer during compression/expansion. This volume consists of the regenerator, heat exchangers, connections, and any space in the cylinders which is not affected by the piston or displacer position.

The porosity, or porosity factor, is the ratio of open-space volume to total geometric volume in the regenerator, i.e., the fraction of empty space in the regenerator.

The calculated heat transfer rate of a regenerator or heat exchanger for an ideal gas passing through is the gas mass flow rate times the specific heat at constant pressure of the gas times the gas temperature change across the heat exchanger. The theoretical maximum heat transfer rate is not achieved due to the heat exchanger finite size, the heat capacity of the heat exchanger itself, and similar factors. These losses are accounted for in the following programs through a heat exchanger efficiency, defined by actual heat transfer rate divided by the maximum theoretical rate.

The sample output files shown are not the complete version of the output files, which also contain values of the various quantities at a number of positions during the cycle.

An important point to note is that the performance of the heat pump depends on the engine frequency, and can vary dramatically for different fluids.

The maximum COP for a Stirling device depends on proper proportioning of the space volumes in the device. The first four of the Stirling programs are used to obtain a first approximation to the volumes of the various spaces for a given thermal load. They can also be used to determine the optimum ratio of the volumes to achieve the maximum possible COP. The last two programs include losses in the various portions of the device, especially the losses which occur in the regenerator. These losses substantially reduce the COP of the device.

K.3.1 STIRL1.FOR

This program is adapted from a QUICKBASIC program given in Appendix A of *Stirling and Vuilleumier Heat Pumps* by J. Wurm, J. A. Kinast, T. R. Roose, and W. R. Staats. The heat pump modelled is a thermal-compression Vuilleumier cycle heat pump as shown in Figure G-4 of Appendix G. This heat pump usually has an axial configuration (the displacer motion is axial rather than radial within the cylinder), with the phase angle between the engine displacer and refrigerator displacer often set at 90°. Pressure ratios and heat capacities are typically low for these heat pumps. Heat exchange occurs with the surroundings, with work transfer between the engine and refrigerator taking place internally by the gas. Since this is a thermal-compression device no mechanical energy storage is required .

The analysis is simplified by assuming isothermal and ideal behavior (no losses). That is, the hot chamber, warm-hot chamber, warm-cold chamber, and cold chamber remain at the temperature set by the operator throughout the analysis. The temperatures of the warm-hot and warm-cold chambers are equal. A total volume and desired mean operating pressure is input by the operator. The analysis takes the volume of each chamber to vary sinusoidally with time. Instantaneous pressure is computed by calculation from the ideal gas law. Each chamber is taken to be instantaneously at the same pressure. The gas mass is computed to meet the total volume/mean pressure constraint. Heat transfer, work into and out of the heat pump and its spaces, and the necessary heat storage required (assuming 100% efficiency) for the regenerator are computed. The heat exchangers are redundant in an isothermal analysis since all the heat transfer can occur across the boundaries of the constant temperature chambers. They do not participate directly in this analysis, although their presence is accounted for through the addition of their volume to the dead space. By altering the volume ratio between the hot/warm cylinder and the warm/cold cylinder, optimum specific capacity can be determined.

Data to be input is the following:

- Drive to be used for output files
- Desired units
- Gas used as refrigerant
- Desired mean operating pressure
- T(hot chamber)
- T(warm chambers)
- T(cold chamber)
- Total volume
- Void volume fraction (ratio of dead space to total volume)
- Ratio of volume fractions (hot/warm spaces)/(warm/cold spaces)
- Number of cylinders

The program computes the following instantaneous quantities at one degree intervals of crankshaft rotation:

- hot, warm-hot, warm-cold, and cold volumes
- pressure, heat and mass transfer.

No net work is produced by this machine since this is a thermal compression Stirling machine.

Sample output for STIRL1.FOR - SI units

```
            Thermal compression Vuilleumier heat pump
    Gas is HELIUM.
  Temperature summary
        Thot = 538.00, Twarm =  65.70, Tcold =    .00 [deg C]
  Volume summary
        Total volume =  1000.00 [cm^3]
        Vhot/Vcold =  .001, dead space volume fraction =  2.000
        degrees    hot   warm/hot  warm/cold   cold
            0      333.    333.      333.       0.  [cm^3]
           90      666.      0.      167.     167.  [cm^3]
          180      333.    333.        0.     333.  [cm^3]
          270        0.    666.      167.     167.  [cm^3]
  Pressure summary
        Pressure: max =  6.33, min =  3.93, avg =  5.00 [MPa]
        Pressure: degrees => 0:  5.09, 90:  6.29, 180:  4.63, 270:  3.94 [MPa]
  Energy transfer summary
        Heat in:  hot 3417.10, w/h  3166.72,  w/c  1371.52, cold  1977.62 [J]
        Heat out: hot -3166.72, w/h -3417.10,  w/c -1977.62, cold -1371.52 [J]
```

```
          Net heat: hot   250.38, w/h  -250.38,  w/c -606.10,   cold   606.10 [J]
     Regenerator storage: hot<=>warm =  6107.5  warm<=>cold =    977.5 [J]
     Mass transfer summary
          Mass transfer hot<=>warm    2.4920 warm<=>cold    2.8673 [kg]
```

Sample output for STIRL1.FOR - English units

```
          Thermal compression Vuilleumier heat pump
   Gas is HELIUM.
     Temperature summary
          Thot =1000.00, Twarm = 150.00, Tcold =  32.00 [deg F]
     Volume summary
          Total volume =    61.02 [in^3]
          Vhot/Vcold =  .001, dead space volume fraction =  2.000
          degrees     hot   warm/hot  warm/cold   cold
             0        20.     20.       20.         0.  [in^3]
            90        41.      0.       10.        10.  [in^3]
           180        20.     20.        0.        20.  [in^3]
           270         0.     41.       10.        10.  [in^3]
     Pressure summary
      Pressure: max = 917.90, min = 569.72, avg = 725.20 [psia]
      Pressure: degrees => 0: 737.92, 90: 912.17, 180: 671.35, 270: 571.95 [psia]
     Energy transfer summary
          Heat in:  hot      3.24, w/h   3.00, w/c    1.30, cold     1.87 [Btu]
          Heat out: hot    -3.00, w/h   -3.24, w/c   -1.87, cold    -1.30 [Btu]
          Net heat: hot      .24,  w/h   -.24, w/c    -.57, cold      .57 [Btu]
     Regenerator storage: hot<=>warm =    5.8  warm<=>cold =      .9 [Btu]
     Mass transfer summary
          Mass transfer hot<=>warm    5.4954 warm<=>cold    6.3217 [lbm]
```

K.3.2 STIRL2.FOR

This program was adapted from four QUICKBASIC programs given in Appendices B and C of *Stirling and Vuilleumier Heat Pumps* by J. Wurm, J. A. Kinast, T. R. Roose, and W. R. Staats. The machines are separated into two classifications based on the physical layout. The piston-piston (alpha) configuration (Figure G-1 of Appendix G) has two separate cylinders, while the piston-displacer (beta) configuration (Figure G-2 of Appendix G) has a single cylinder. These heat pumps have axial configurations, with the phase angle between the piston and the displacer/piston usually set at 90°. Heat exchange occurs with the surroundings, with work transfer between the engine and refrigerator taking place through the power piston (work transfer cannot occur by the gas because the fluid in the engine is separate from the fluid in the refrigerator). Mechanical energy storage is required because this is a mechanical-compression device.

In this program Stirling engines and refrigerators can either be analyzed individually or in a combined heat pump configuration. The refrigerator section alone would require work input from some external source (not necessarily mechanically connected, a solenoid could drive the piston). In the heat pump analysis, the refrigerator portion is run first to determine the net work required; then the engine portion is run to determine the engine size required to produce the necessary work.

The analysis is simplified by assuming ideal behavior (no losses) and isothermal behavior. By adjusting the volume ratio between the cold space and the total volume the device is found to produce or consume net work in addition to pumping heat. At one specific value of this ratio, only heat pumping occurs and there is no net work.

The operator is first asked to choose between a (1) piston-piston and a (2) displacer-piston configuration. The next choice is between (1) an engine alone, (2) a refrigerator alone, and (3) an engine-refrigerator combination.

The data to be input is the following:

- Drive to be used for output files
- Desired units
- Number of cylinders
- Configuration - Piston-piston or displacer-piston
  - Engine alone, refrigerator alone, or engine-refrigerator combination
- Gas used as refrigerant
- For the refrigerator:
  - Desired mean operating pressure
  - Temperatures of warm and cold chambers
  - Dead space cold/warm volume fraction
  - Estimate of total cold/warm volume
- For the engine:
  - Desired mean operating pressure
  - Temperature of hot chamber
  - Dead space volume fraction

The program iterates until volumes are found which meet the parameters. The program computes the following instantaneous quantities at one degree intervals of crankshaft rotation:

hot, warm-hot, warm-cold, and cold volumes
pressure, heat and mass transfer, work performed.

Sample output for STIRL2.FOR - SI units

STIRLING ENGINE/REFRIGERATOR - Piston-Piston
Gas is HELIUM.

Stirling piston-piston refrigerator
Temperature summary
Twarm = 65.70, Tcold = .00 [deg C]
Volume summary
Total volume = 3168.30 [cm^3], dead space volume fraction = .400
Volumes: max = 2256.25, min = 912.05
0 degrees: vol 1 = 792.08 vol 2 = 1267.32 [cm^3]
45 degrees: vol 1 = 456.03 vol 2 = 1128.12 [cm^3]
90 degrees: vol 1 = 316.83 vol 2 = 792.08 [cm^3]
135 degrees: vol 1 = 456.03 vol 2 = 456.03 [cm^3]

180 degrees: vol 1 = 792.07 vol 2 = 316.83 [cm^3]
225 degrees: vol 1 = 1128.12 vol 2 = 456.03 [cm^3]
270 degrees: vol 1 = 1267.32 vol 2 = 792.07 [cm^3]
315 degrees: vol 1 = 1128.12 vol 2 =1128.12 [cm^3]

Pressure summary

Pressure: degrees => 0: 3.57, 90: 6.77 [MPa]
180: 6.18, 270: 3.39 [MPa]
Pressure: max = 7.89, min = 3.17, avg = 5.00 [MPa]

Energy transfer summary

0 degrees, heat 1: -29.652, 2: -.259 [J]
90 degrees, heat 1: .492, 2: -56.360 [J]
180 degrees, heat 1: 51.021, 2: .445 [J]
270 degrees, heat 1: -.245, 2: 28.105 [J]
Heat in: 1 = 5769.86, 2 = 3485.47 [J]
Heat out: 1 = -3670.48, 2 = -6089.80 [J]
Net heat: 1 = 2099.38, 2 = -2604.33 [J]

Regenerator storage: (1-2) = -1804.7190 [J]
Work out = 6246.6, work in = -6751.6, work absorbed = -505.0 [J]

Mass transfer summary

Mass transfer in/out: (1-2): 5.293 [kg]

****

Stirling piston-piston engine

Temperature summary

Thot = 538.00, Twarm = 65.70 [deg C]

Volume summary

Total volume = 821.31 [cm^3],
dead space volume fraction = .400
Volumes: max = 584.88, min = 236.43
0 degrees: vol 1 = 205.33 vol 2 = 328.52 [cm^3]
45 degrees: vol 1 = 118.21 vol 2 = 292.44 [cm^3]
90 degrees: vol 1 = 82.13 vol 2 = 205.33 [cm^3]
135 degrees: vol 1 = 118.21 vol 2 = 118.21 [cm^3]
180 degrees: vol 1 = 205.33 vol 2 = 82.13 [cm^3]
225 degrees: vol 1 = 292.44 vol 2 = 118.21 [cm^3]
270 degrees: vol 1 = 328.52 vol 2 = 205.33 [cm^3]
315 degrees: vol 1 = 292.44 vol 2 = 292.44 [cm^3]

Pressure summary

Pressure: degrees => 0: 3.12, 90: 5.40 [MPa]
180: 7.70, 270: 3.78 [MPa]
Pressure: max = 8.21, min = 3.05, avg = 5.00 [MPa]

Energy transfer summary

0 degrees, heat 1: -6.720, 2: -.059 [J]
90 degrees, heat 1: .102, 2: -11.657 [J]
180 degrees, heat 1: 16.517, 2: .144 [J]
270 degrees, heat 1: -.071, 2: 8.093 [J]
Heat in: 1 = 1699.62, 2 = 1016.81 [J]
Heat out: 1 = -832.38, 2 = -1379.09 [J]
Net heat: 1 = 867.24, 2 = -362.28 [J]

Regenerator storage: (1-2) = 1821.5460 [J]
Work out = 1946.0, work in = -1441.1, net work = 505.0 [J]

Mass transfer summary

Mass transfer in/out: (1-2): .743 [kg]

Sample output for STIRL2.FOR - English units

STIRLING ENGINE/REFRIGERATOR - Piston-Piston

Gas is HELIUM.

Stirling piston-piston refrigerator

Temperature summary

Twarm = 150.00, Tcold = 32.00 [deg F]

Volume summary

Total volume = 193.34 [in^3],

dead space volume fraction = .400

Volumes: max = 137.68, min = 55.66

0 degrees: vol 1 = 48.33 vol 2 = 77.34 [in^3]

45 degrees: vol 1 = 27.83 vol 2 = 68.84 [in^3]

90 degrees: vol 1 = 19.33 vol 2 = 48.33 [in^3]

135 degrees: vol 1 = 27.83 vol 2 = 27.83 [in^3]

180 degrees: vol 1 = 48.33 vol 2 = 19.33 [in^3]

225 degrees: vol 1 = 68.84 vol 2 = 27.83 [in^3]

270 degrees: vol 1 = 77.34 vol 2 = 48.33 [in^3]

315 degrees: vol 1 = 68.84 vol 2 = 68.84 [in^3]

Pressure summary

Pressure: degrees => 0: 517.30, 90: 982.05 [psia]

180: 895.82, 270: 492.34 [psia]

Pressure: max =1144.01, min = 459.71, avg = 725.20 [psia]

Energy transfer summary

0 degrees, heat 1: -.028, 2: .000 [Btu]

90 degrees, heat 1: .000, 2: -.053 [Btu]

180 degrees, heat 1: .048, 2: .000 [Btu]

270 degrees, heat 1: .000, 2: .027 [Btu]

Heat in: 1 = 5.47, 2 = 3.30 [Btu]

Heat out: 1 = -3.48, 2 = -5.77 [Btu]

Net heat: 1 = 1.99, 2 = -2.47 [Btu]

Regenerator storage: (1-2) = -1.7072 [Btu]

Work out = 5.9, work in = -6.4, work absorbed = -.5 [Btu]

Mass transfer summary

Mass transfer in/out: (1-2): 11.673 [lbm]

****

Stirling piston-piston engine

Temperature summary

Thot = 1000.00, Twarm = 150.00 [deg F]

Volume summary

Total volume = 50.01 [in^3],

dead space volume fraction = .400

Volumes: max = 35.62, min = 14.40

0 degrees: vol 1 = 12.50 vol 2 = 20.01 [in^3]

45 degrees: vol 1 = 7.20 vol 2 = 17.81 [in^3]

90 degrees: vol 1 = 5.00 vol 2 = 12.50 [in^3]

135 degrees: vol 1 = 7.20 vol 2 = 7.20 [in^3]

180 degrees: vol 1 = 12.50 vol 2 = 5.00 [in^3]

225 degrees: vol 1 = 17.81 vol 2 = 7.20 [in^3]

270 degrees: vol 1 = 20.01 vol 2 = 12.50 [in^3]

315 degrees: vol 1 = 17.81 vol 2 = 17.81 [in^3]

Pressure summary

Pressure: degrees => 0: 452.79, 90: 782.80 [psia]

180:1117.29, 270: 547.62 [psia]

Pressure: max =1190.40, min = 441.79, avg = 725.20 [psia]
Energy transfer summary
0 degrees, heat 1: -.006, 2: .000 [Btu]
90 degrees, heat 1: .000, 2: -.011 [Btu]
180 degrees, heat 1: .016, 2: .000 [Btu]
270 degrees, heat 1: .000, 2: .008 [Btu]
Heat in: 1 = 1.61, 2 = .96 [Btu]
Heat out: 1 = -.79, 2 = -1.30 [Btu]
Net heat: 1 = .82, 2 = -.34 [Btu]
Regenerator storage: (1-2) = 1.7231 [Btu]
Work out = 1.8, work in = -1.4, net work = .5 [Btu]
Mass transfer summary
Mass transfer in/out: (1-2): 1.636 [lbm]

Sample output for STIRL2.FOR - SI units

STIRLING ENGINE/REFRIGERATOR - Displacer-Piston

Gas is HELIUM.

Stirling displacer-piston refrigerator

Temperature summary
Twarm = 65.70, Tcold = .00 [deg C]
Volume summary
Total volume = 1000.00 [cm^3], dead
space volume fraction = .001
Offset volume = .000 [cm^3]
Volumes: warm = 499.5, cold = 499.5 [cm^3]
Volumes: max = 999.71, min = 414.85 [cm^3]
Volumes: min hot = .8, min warm = .5 [cm^3]
Volumes: max hot = 585.6, max cold = 827.6 [cm^3]
0 degrees: vol 1 = 585.65 vol 2 = 121.63 [cm^3]
45 degrees: vol 1 = 500.00 vol 2 = 414.06 [cm^3]
90 degrees: vol 1 = 293.22 vol 2 = 706.49 [cm^3]
135 degrees: vol 1 = 86.44 vol 2 = 827.61 [cm^3]
180 degrees: vol 1 = .79 vol 2 = 706.49 [cm^3]
225 degrees: vol 1 = 86.44 vol 2 = 414.06 [cm^3]
270 degrees: vol 1 = 293.22 vol 2 = 121.63 [cm^3]
315 degrees: vol 1 = 500.00 vol 2 = .50 [cm^3]
Pressure summary
Pressure: degrees => 0: 4.23, 90: 3.35 [MPa]
180: 5.07, 270: 7.39 [MPa]
Pressure: max = 7.52, min = 3.33, avg = 5.00 [MPa]
Energy transfer summary
0 degrees, heat 1: -.187, 2: 21.704 [J]
90 degrees, heat 1: -17.113, 2: 16.964 [J]
180 degrees, heat 1: .227, 2: -26.193 [J]
270 degrees, heat 1: 37.662, 2: -37.333 [J]
Heat in: 1 = 3892.56, 2 = 3072.33 [J]
Heat out: 1 = -2095.77, 2 = -5301.30 [J]
Net heat: 1 = 1796.79, 2 = -2228.97 [J]
Regenerator storage: (1-2) = -1733.3600 [J]
Work out = 2637.6, work in = -3069.8, work absorbed = -432.2 [J]
Mass transfer summary
Mass transfer in/out: (1-2): 5.084 [kg]
****

Stirling displacer-piston engine

Temperature summary

Thot = 538.00, Twarm = 65.70 [deg C]

Volume summary

Total volume = 263.29 [cm^3],

dead space volume fraction = .001

Offset volume = .000 [cm^3], hot = 131.5, warm = 131.5 [cm^3]

Volumes: max = 263.21, min = 109.23 [cm^3]

Volumes: min hot = .2, min warm = .1 [cm^3]

Volumes: max hot = 154.2, max warm = 217.9 [cm^3]

0 degrees: vol 1 = 154.20 vol 2 = 32.02 [cm^3]

45 degrees: vol 1 = 131.65 vol 2 = 109.02 [cm^3]

90 degrees: vol 1 = 77.20 vol 2 = 186.01 [cm^3]

135 degrees: vol 1 = 22.76 vol 2 = 217.90 [cm^3]

180 degrees: vol 1 = .21 vol 2 = 186.01 [cm^3]

225 degrees: vol 1 = 22.76 vol 2 = 109.02 [cm^3]

270 degrees: vol 1 = 77.20 vol 2 = 32.02 [cm^3]

315 degrees: vol 1 = 131.65 vol 2 = .13 [cm^3]

Pressure summary

Pressure: degrees => 0: 5.68, 90: 2.51 [MPa]

180: 2.95, 270: 8.53 [MPa]

Pressure: max = 10.51, min = 2.38, avg = 5.00 [MPa]

Energy transfer summary

0 degrees, heat 1: -.066, 2: 7.651 [J]

90 degrees, heat 1: -3.369, 2: 3.339 [J]

180 degrees, heat 1: .035, 2: -4.007 [J]

270 degrees, heat 1: 11.529, 2: -11.429 [J]

Heat in: 1 = 1173.76, 2 = 853.06 [J]

Heat out: 1 = -431.52, 2 = -1163.12 [J]

Net heat: 1 = 742.24, 2 = -310.06 [J]

Regenerator storage: (1-2) = 1903.2030 [J]

Work out = 955.0, work in = -522.8, net work = 432.2 [J]

Mass transfer summary

Mass transfer in/out: (1-2): .777 [kg]

Sample output for STIRL2.FOR- English units

STIRLING ENGINE/REFRIGERATOR - Displacer-Piston

Gas is HELIUM.

Stirling displacer-piston refrigerator

Temperature summary

Twarm = 150.00, Tcold = 32.00 [deg F]

Volume summary

Total volume = 61.02 [in^3],

dead space volume fraction = .001

Offset volume = .000 [in^3]

Volumes: warm = 30.5, cold = 30.5 [in^3]

Volumes: max = 61.00, min = 25.31 [in^3]

Volumes: min hot = .0, min warm = .0 [in^3]

Volumes: max hot = 35.7, max cold = 50.5 [in^3]

0 degrees: vol 1 = 35.74 vol 2 = 7.42 [in^3]

45 degrees: vol 1 = 30.51 vol 2 = 25.27 [in^3]

90 degrees: vol 1 = 17.89 vol 2 = 43.11 [in^3]

135 degrees: vol 1 = 5.27 vol 2 = 50.50 [in^3]

180 degrees: vol 1 = .05 vol 2 = 43.11 [in^3]
225 degrees: vol 1 = 5.27 vol 2 = 25.27 [in^3]
270 degrees: vol 1 = 17.89 vol 2 = 7.42 [in^3]
315 degrees: vol 1 = 30.51 vol 2 = .03 [in^3]

Pressure summary

Pressure: degrees => 0: 613.42, 90: 486.02 [psia]
180: 735.12, 270:1071.83 [psia]
Pressure: max =1090.47, min = 482.28, avg = 725.20 [psia]

Energy transfer summary

0 degrees, heat 1: .000, 2: .021 [Btu]
90 degrees, heat 1: -.016, 2: .016 [Btu]
180 degrees, heat 1: .000, 2: -.025 [Btu]
270 degrees, heat 1: .036, 2: -.035 [Btu]
Heat in: 1 = 3.69, 2 = 2.91 [Btu]
Heat out: 1 = -1.99, 2 = -5.02 [Btu]
Net heat: 1 = 1.70, 2 = -2.11 [Btu]

Regenerator storage: (1-2) = -1.6396 [Btu]
Work out = 2.5, work in = -2.9, work absorbed = -.4 [Btu]

Mass transfer summary

Mass transfer in/out: (1-2): 11.211 [lbm]

****

Stirling displacer-piston engine

Temperature summary

Thot = 1000.00, Twarm = 150.00 [deg F]

Volume summary

Total volume = 16.03 [in^3],
dead space volume fraction = .001
Offset volume = .000 [in^3], hot = 8.0, warm = 8.0 [in^3]
Volumes: max = 16.03, min = 6.65 [in^3]
Volumes: min hot = .0, min warm = .0 [in^3]
Volumes: max hot = 9.4, max warm = 13.3 [in^3]
0 degrees: vol 1 = 9.39 vol 2 = 1.95 [in^3]
45 degrees: vol 1 = 8.02 vol 2 = 6.64 [in^3]
90 degrees: vol 1 = 4.70 vol 2 = 11.33 [in^3]
135 degrees: vol 1 = 1.39 vol 2 = 13.27 [in^3]
180 degrees: vol 1 = .01 vol 2 = 11.33 [in^3]
225 degrees: vol 1 = 1.39 vol 2 = 6.64 [in^3]
270 degrees: vol 1 = 4.70 vol 2 = 1.95 [in^3]
315 degrees: vol 1 = 8.02 vol 2 = .01 [in^3]

Pressure summary

Pressure: degrees => 0: 824.43, 90: 364.24 [psia]
180: 427.18, 270:1236.95 [psia]
Pressure: max =1523.92, min = 345.10, avg = 725.20 [psia]

Energy transfer summary

0 degrees, heat 1: .000, 2: .007 [Btu]
90 degrees, heat 1: -.003, 2: .003 [Btu]
180 degrees, heat 1: .000, 2: -.004 [Btu]
270 degrees, heat 1: .011, 2: -.011 [Btu]
Heat in: 1 = 1.11, 2 = .81 [Btu]
Heat out: 1 = -.41, 2 = -1.10 [Btu]
Net heat: 1 = .70, 2 = -.29 [Btu]

Regenerator storage: (1-2) = 1.8002 [Btu]
Work out = .9, work in = -.5, net work = .4 [Btu]

Mass transfer summary

Mass transfer in/out: (1-2): 1.709 [lbm]

K.3.3 STIRL3.FOR

This program was adapted from a QUICKBASIC program given in Appendix D of *Stirling and Vuilleumier Heat Pumps* by J. Wurm, J. A. Kinast, T. R. Roose, and W. R. Staats. The heat pump modelled is a mechanical-compression Vuilleumier heat pump as shown in Figure G-5 of Appendix G. The phase angle between the pistons is often taken as 90°. Heat exchange occurs with the surroundings, with work transfer between the engine and refrigerator taking place internally in the gas. Mechanical energy storage is required since this is a mechanical-compression device.

The analysis is simplified by assuming ideal behavior (no losses) and isothermal behavior.

Data to be input is the following:

- Drive to be used for output files
- Desired units
- Gas used as refrigerant
- Desired mean operating pressure
- Temperatures of hot, warm, and cold chambers
- Total volume
- Dead space volume fraction
- Ratio of the cold volume to total volume
- Number of cylinders

The program computes the following instantaneous quantities at one degree intervals of crankshaft rotation:

- hot, warm-hot, warm-cold, and cold volumes
- pressure, heat and mass transfer, work performed.

Sample output for STIRL3.FOR - SI units

```
              Mechanical compression Vuilleumier heat pump
   Gas is HELIUM.
   Thot =  538.0, Twarm =  65.7, Tcold =    .0   [deg C]
   Volume summary
          Total  1000.0 Vmax =  1166.23, Vmin =   125.57 [cm^3]
          cold vol./total =   70.82 %, dead space =  .001
          Degrees      hot      warm       cold
              0      145.90       .15    354.10 [cm^3]
             90      291.65    145.90    707.85 [cm^3]
            180      145.90    291.65    354.10 [cm^3]
            270         .15    145.90       .35 [cm^3]
            360      145.90       .15    354.10 [cm^3]
   Pressure: degrees   0 =   3.82,    90 =   1.67 [MPa]
                     180 =   2.42,  270 =   13.07 [MPa]
   Pressure:  max =  15.24,  min =    1.64,  avg =   5.00 [MPa]
  Energy transfer summary
```

```
         90 Heat hot =     .04, warm =    4.25, cold =      .09 [J]
        180 Heat hot =   -6.11, warm =     .05, cold =   -14.84 [J]
        270 Heat hot =    -.29, warm =  -32.96, cold =     -.70 [J]
        Heat in:  hot = 1567.16, warm =  549.83, cold =  3803.51 [J]
        Heat out: hot = -918.52, warm = -2773.70, cold = -2229.26 [J]
        Net heat: hot =  648.64, warm = -2223.87, cold =  1574.25 [J]
        Net heat:      -.974   [J]
Regenerator storage  (hot-warm) =    808.93 [J]
  Regenerator storage  (warm-cold) =   811.01 [J]
  Work storage,  4280.8   [J]
  Mass transfer summary
        Mass transfer in/out (hot-warm) =     .330 [kg]
        Mass transfer in/out (warm-cold) =   2.379 [kg]
```

Sample output for STIRL3.FOR - English units

```
        Mechanical compression Vuilleumier heat pump
  Gas is HELIUM.
  Thot = 1000.0, Twarm =  150.0, Tcold =  32.0   [deg F]
  Volume summary
        Total   61.0 Vmax =   71.16, Vmin =    7.66 [in^3]
        Cold vol./total =  70.82 %, dead space =  .001
        Degrees     hot     warm     cold
          0        8.90      .01    21.61 [in^3]
         90       17.80     8.90    43.19 [in^3]
        180        8.90    17.80    21.61 [in^3]
        270         .01     8.90      .02 [in^3]
        360        8.90      .01    21.61 [in^3]
        Pressure: degrees  0 =  554.86,   90 =  242.29 [psia]
                         180 =  350.55, 270 = 1895.66 [psia]
        Pressure:  max = 2210.02,  min =  237.96,   avg =  725.20 [psia]
  Energy transfer summary
         90 Heat hot =     .00, warm =     .00, cold =     .00 [Btu]
        180 Heat hot =    -.01, warm =     .00, cold =    -.01 [Btu]
        270 Heat hot =     .00, warm =    -.03, cold =     .00 [Btu]
        Heat in:  hot =   1.49, warm =     .52, cold =    3.61 [Btu]
        Heat out:  hot =   -.87, warm =   -2.63, cold =   -2.11 [Btu]
        Net heat:  hot =    .61, warm =   -2.11, cold =    1.49 [Btu]
        Net heat:       .000   [Btu]
  Regenerator storage  (hot-warm) =      .77 [Btu]
  Regenerator storage  (warm-cold) =     .77 [Btu]
  Work storage,     4.1   [Btu]
  Mass transfer summary
        Mass transfer in/out (hot-warm) =      .728 [lbm]
        Mass transfer in/out (warm-cold) =    5.245 [lbm]
```

K.3.4 STIRL4.FOR

This program was adapted from a QUICKBASIC program given in Appendix E of *Stirling and Vuilleumier Heat Pumps* by J. Wurm, J. A. Kinast, T. R. Roose, and W. R. Staats. The program models an Ericsson-Ericsson heat pump as shown in Figure G-6 of Appendix G. This is thermodynamically similar to the mechanical compression Vuilleumier heat pump because the working fluid can occupy any of the chambers. The Ericsson-Ericsson heat pump differs from the Vuilleumier heat pump in that the engine displacer and the refrigerator displacer move in phase with one another, and pressure variations are caused by the relative motion between the two flywheel pistons. Heat exchange occurs with the surroundings, with work transfer between the engine and refrigerator taking place internally by the gas. Mechanical energy storage is required since this is a mechanical-compression device.

The analysis is simplified by assuming ideal behavior (no losses) and isothermal behavior. By adjusting the volume ratio between the cold space and the hot space, net work can be produced or consumed. An optimum value of the ratio is when the net work is zero. Also, the volume ratio between the warm space and the total volume determines the degree of mechanical compression, which affects specific capacity. The optimum value results in the highest specific capacity.

Data to be input is the following:

- Drive to be used for output files
- Desired units
- Gas used as refrigerant
- Desired mean operating pressure
- Temperature of hot, warm, and cold chambers
- Total volume
- Dead space volume fraction
- Ratio of the fractions (volume fraction between flywheel pistons)/(ratio of cold space volume to total volume)
- Ratio of cold to hot volumes
- Number of cylinders

The program computes the following instantaneous quantities at one degree intervals of crankshaft rotation:

- hot, warm-hot, warm-cold, and cold volumes
- pressure, heat and mass transfer, work performed.

Sample output for STIRL4.FOR - SI units

```
                Ericsson/Ericsson Heat Pump
        Gas is HELIUM.
        Temperature summary
                Thot =  538.00, Twarm =  65.70, Tcold=   .00 [deg C]
        Volume summary
                Total volume =1000.000, dead space= .001 [cm^3]
                Fraction of space - flywheel pistons to total volume =  .6200
                Volume ratio - cold space to hot space =    2.427
                Vmax =  999.69, Vmin =  380.31 [cm^3]
                  0 degrees, Volume: hot =    55.4, warm =  190.3, cold =  134.6 [cm^3]
                 90 degrees, Volume: hot =   110.8, warm =  310.2, cold =  269.0 [cm^3]
                180 degrees, Volume: hot =    55.4, warm =  809.7, cold =  134.6 [cm^3]
                270 degrees, Volume: hot =      .1, warm =  689.8, cold =     .1 [cm^3]
        Pressure summary
```

Pressure: degrees => 0: 8.11, 90: 4.47 [MPa]
180: 3.08, 270: 4.47 [MPa]
Pressure: max= 8.11, min= 3.08, avg= 5.00 [MPa]

Energy transfer summary

90 degrees, heat 1: .04, 2: 24.11, 3: .09 [J]
180 degrees, heat 1: -2.98, 2: 10.36, 3: -7.24 [J]
270 degrees, heat 1: -.04, 2: -23.93, 3: -.09 [J]
360 degrees, heat 1: 7.83, 2: -27.23, 3: 19.02 [J]
Heat in: h: 780.45, w: 2864.57, c: 1894.47 [J]
Heat out: h: -368.16, w: -4278.32, c: -893.67 [J]
Net heat: h: 412.29, w: -1413.75, c: 1000.80 [J]
Net heat: -.66 [J]

Regenerator storage: (h-w): 901.45, (w-c): 903.92 [J]
Work storage: 2979.6 [J]
Mass transfer summary

Mass transfer in/out: (h-w): .368, (w-c): 2.651 [kg]

Sample output for STIRL4.FOR - English units

Ericsson/Ericsson Heat Pump

Gas is HELIUM.

Temperature summary

Thot = 1000.00, Twarm = 150.00, Tcold= 32.00 [deg F]

Volume summary

Total volume = 61.020, dead space= .001 [in^3]
Fraction of space - flywheel pistons to total volume = .6200
Volume ratio - cold space to hot space = 2.427
Vmax = 61.00, Vmin = 23.21 [in^3]
0 degrees, Volume: hot = 3.4, warm = 11.6, cold = 8.2 [in^3]
90 degrees, Volume: hot = 6.8, warm = 18.9, cold = 16.4 [in^3]
180 degrees, Volume: hot = 3.4, warm = 49.4, cold = 8.2 [in^3]
270 degrees, Volume: hot = .0, warm = 42.1, cold = .0 [in^3]

Pressure summary

Pressure: degrees => 0:1175.75, 90: 648.04 [psia]
180: 447.30, 270: 648.07 [psia]
Pressure: max=1175.75, min= 447.30, avg= 725.20 [psia]

Energy transfer summary

90 degrees, heat 1: .00, 2: .02, 3: .00 [Btu]
180 degrees, heat 1: .00, 2: .01, 3: -.01 [Btu]
270 degrees, heat 1: .00, 2: -.02, 3: .00 [Btu]
360 degrees, heat 1: .01, 2: -.03, 3: .02 [Btu]
Heat in: h: .74, w: 2.72, c: 1.80 [Btu]
Heat out: h: -.35, w: -4.05, c: -.85 [Btu]
Net heat: h: .39, w: -1.34, c: .95 [Btu]
Net heat: .00 [Btu]

Regenerator storage: (h-w): .85, (w-c): .85 [Btu]
Work storage: 2.8 [Btu]
Mass transfer summary

Mass transfer in/out: (h-w): .811, (w-c): 5.846 [lbm]

K.3.5 STIRL5.FOR

This program was adapted from a QUICKBASIC program given in Appendix F of *Stirling and Vuilleumier Heat Pumps* by J. Wurm, J. A. Kinast, T. R. Roose, and W. R. Staats. It is an elaboration of the previous four programs in the sense that, while the thermodynamic simulation is much the same, in this program energy losses in the heat exchangers and regenerators are now included. Inclusion of these factors results in a mathematically nonlinear problem which is solved by iteration. In the initial step the void volume space is taken as zero, and the mass, mass transfer, heat transfer, and energy transfer are all calculated. From this information cooling capacity and cooling coefficient of performance (COP) are calculated. Then associated void volumes and friction losses are calculated, added to the total volume and energy balance, and the process is repeated. At the end of each iteration the newly-calculated cooling COP is compared with the previously calculated one, and the calculation is repeated until the change in COP is less than a fixed number (1 %). At this point the calculated cooling capacity is compared with the desired cooling capacity. If these do not agree to within 1 % the hot space volume is adjusted and the process repeated.

Data to be input is the following:

- Drive to be used for output files
- Desired units
- Gas used as refrigerant
- Target cooling capacity
- Desired mean operating pressure
- Temperatures of hot, hot/warm, cold/warm, and cold chambers
- Estimate of expected hot space volume
- Engine speed
- Ratio of cold space volume to hot space volume
- Hot and cold regenerator properties
  - Matrix porosity, mass density, surface density, length-to-diameter ratio of hot and cold regenerators, efficiencies of hot and cold regenerators.
- Hot, hot-warm, cold-warm, cold heat exchanger properties
  - Differential temperatures for hot, hot/warm, cold/warm, and hot heat exchangers, Reynolds numbers for hot, hot/warm, cold/warm, and cold exchangers, ratio of free flow area to frontal area, ratio of heat transfer area to total volume, flow passage hydraulic diameter, fin thickness, height, thermal conductivity, ratio of fin area to total fin volume, porosity.

When all of the desired power, pressure, and temperature conditions are met the program computes the following instantaneous quantities at one degree intervals of crankshaft rotation:

hot, warm-hot, warm-cold, and cold volumes
pressure, heat and mass transfer, work performed.

Sample output for STIRL5.FOR - SI units

```
               THERMAL COMPRESSION VUILLEUMIER
        ------------ cycle parameters ------------
 Gas is Helium.
 Target ideal cooling capacity                         35.00  [kW]
 Hot side heat input temperature                       547.8  [deg C]
 Hot side heat rejection temperature                    65.7  [deg C]
 Cold side heat rejection temp.                         65.7  [deg C]
 Cold side heat input temperature                         .0  [deg C]
 Hot space volume                                     7378.8  [cm^3]
 Cold space volume                                    5666.4  [cm^3]
 Void volume percentage 71.3%,   void volume         32415.7  [cm^3]
 Total volume                                        45461.0  [cm^3]
 Mean pressure                                          5.00  [MPa]
 Maximum pressure     5.25  minimum pressure            4.75  [MPa]
 Pressure ratio      1.11    mean pressure              5.00  [MPa]
 Gas mass       321.9 [gm]    cycle frequency        1000. [rpm]
        ----------------- ideal cycle --- real cycle ---
 Cooling COP                 2.441       .274
 Cooling EER                 8.331       .936
 Heat input        [kW]     14.315     62.562
 Cooling capacity  [kW]     34.951     17.163
 Total fric. losses [kW]               17.905
        --------------- regenerators ---------------
 Porosity                                              .80
 Regenerator density                             1538.00 [kg/m^3]
 Regenerator surface density                   9.50 [m^2/kg]
 Regenerator surface area/volume ratio          14611.00 [1/m]
        -------------------- hot ------- cold ------
 Reynolds number            221.495      469.178
 Diameter         [mm]      197.751      277.208
 Length           [mm]       39.550       55.442
 Volume         [cm^3]     1214.712     3346.090
 Length/diameter ratio         .200         .200
 Void volume    [cm^3]      971.769     2676.872
 Efficiency        [%]       95.001       95.001
 Heat exchg reg/gas [kW]    965.031      355.791
 Heat leak        [kW]       48.247       17.788
 Gas temp. change  [deg C]   24.103        3.285
 Pressure drop     [kPa]     31.398       46.999
 Flow fric. power   [kW]      5.066       12.274
 Total regenerator flow friction losses  [kW]   17.340
        --------------- heat exchangers ---------------
 Material is  Custom fin material
 Material thermal conductivity is        173.06 [W/m-K]
 Configuration is finned tube, cross-flow.
 Heat transfer area/total volume         587.00 [m^2/m^3]
 Flow passage hydraulic diameter          3.632 [mm]
 Fin thickness                            .3302 [mm]
 Fin heights                             7.5946 [mm]
 Fin area/total area ratio                 .913
 Ht. exch. free flow area/total frontal area of ht. exch,   .534
 Porosity of heat exchanger geometry                     .76478
```

	hot	hot-warm	cold	cold-warm
Reynolds number	5000.000	8000.000	8000.000	8000.000
Prandtl number	.657	.668	.672	.669
Ht. ex. diff. temp.[deg C]	30.000	6.000	6.000	6.000
Frontal area [cm^2]	272.846	289.725	941.710	826.374
Length [cm]	8.319	16.231	16.888	17.836
Volume [cm^3]	2269.791	4702.564	15903.490	14739.040
Void volume [cm^3]	1735.891	3596.427	12162.670	11272.120
Heat capacity [kW]	62.56	62.56	17.16	17.16
Overall efficiency [%]	43.135	48.439	52.538	49.611
Pressure drop [kPa]	.932	.604	.349	.592
Flow fric. power [kW]	.242	.068	.083	.173
Total heat exchanger flow friction losses [kW]				.566

Sample output for STIRL5.FOR - English units

THERMAL COMPRESSION VUILLEUMIER

— cycle parameters —

Gas is Helium.

Target ideal cooling capacity	119430.00 [Btu/hr]
Hot side heat input temperature	1018.0 [deg F]
Hot side heat rejection temperature	150.0 [deg F]
Cold side heat rejection temp.	150.0 [deg F]
Cold side heat input temperature	32.0 [deg F]
Hot space volume	451.0 [in^3]
Cold space volume	345.4 [in^3]
Void volume percentage 71.3%, void volume	1979.0 [in^3]
Total volume	2775.4 [in^3]
Mean pressure	725.20 [psia]

Maximum pressure 759.30 minimum pressure 687.22 [psia]
Pressure ratio 1.10 mean pressure 722.36 [psia]
Gas mass .7 [lbm] cycle frequency 1000. [rpm]

	ideal cycle	real cycle
Cooling COP	2.448	.276
Cooling EER	8.351	.941
Heat input [Btu/hr]	48589.660	212525.700
Cooling capacity [Btu/hr]	118926.300	58626.000
Total fric. losses [Btu/hr]		60855.150

— regenerators —

Porosity	.80
Regenerator density	96.01 [lbm/ft^3]
Regenerator surface density	46.38 [ft^2/lbm]
Regenerator surface area/volume ratio	4453.13 [1/ft]

	hot	cold
Reynolds number	220.981	468.137
Diameter [in]	7.778	10.904
Length [in]	1.556	2.181
Volume [in^3]	73.910	203.622
Length/diameter ratio	.200	.200
Void volume [in^3]	59.128	162.898
Efficiency [%]	95.001	95.001
Heat exchg reg/gas [Btu/hr]	960.996	353.481
Heat leak [Btu/hr]	48.045	17.672
Gas temp. change [deg F]	43.396	5.899
Pressure drop [psia]	4.540	6.798
Flow fric. power [Btu/hr]	17206.990	41707.520

Total regenerator flow friction losses [Btu/hr] 58914.520

———————— heat exchangers ————————

Material is Custom fin material	
Material thermal conductivity is	100.00 [Btu/hr-ft-R]
Configuration is finned tube, cross-flow.	
Heat transfer area/total volume	178.90 [ft^2/ft^3]
Flow passage hydraulic diameter	.143 [in]
Fin thickness	.0130 [in]
Fin heights	.3000 [in]
Fin area/total area ratio	.913
Ht. exch. free flow area/total frontal area of ht. exch.	.534
Porosity of heat exchanger geometry	.76478

	hot	hot-warm	cold	cold-warm
Reynolds number	5000.000	8000.000	8000.000	8000.000
Prandtl number	.657	.668	.672	.669
Ht. ex. diff. temp.[deg F]	54.000	10.800	10.800	10.800
Frontal area [in^2]	42.107	44.722	145.348	127.579
Length [in]	3.284	6.407	6.686	7.061
Volume [in^3]	138.299	286.550	971.729	900.831
Void volume [in^3]	105.768	219.148	743.159	688.938
Heat capacity [Btu/hr]	212525.70	212525.70	58626.00	58626.00
Overall efficiency [%]	43.012	48.306	52.396	49.476
Pressure drop [psia]	.136	.088	.051	.087
Flow fric. power [Btu/hr]	828.176	232.395	285.889	594.175

Total heat exchanger flow friction losses [Btu/hr] 1940.635

K.3.6 STIRL6.FOR

This program is adapted from a FORTRAN IV program given in the appendix of *Stirling Cycle Engine Analysis* by I. Urieli and D. M. Berchowitz. It can be used to analyze either an engine or a refrigerator. To analyze a heat pump it is necessary to first analyze the refrigerator, then use the work determined to size the engine.

Three different levels of analysis are possible, ideal isothermal, ideal adiabatic, and quasi-steady flow. The cycle diagrams for these three models are given in Figures G-9 through G-14 of Appendix G. Figure G-3 shoould be consulted to see how the "cooler" and "heater" output results differ between an engine and a refrigerator. The ideal isothermal model divides the engine or refrigerator into 5 serially connected compartments, a compression space (c), a "cooler" heat exchanger (k), a regenerator (r), a "heater" heat exchanger (h), and an expansion space (e). In each chamber all of the gas assumed to be at the same temperature and pressure. The gas is assumed to obey the ideal gas law and its total mass in the 5 chambers remains constant during the analysis. Kinetic and potential energies of the gases are neglected, and cyclical steady state is assumed. The thermodynamics used is much the same as in STIRLNG1 through STIRLNG4. Volume variation is determined by varying the compression and expansion volumes according to the drive selected. The various parameters are computed from the equations

M = total mass = $m_c + m_k + m_r + m_h + m_e,$
p = pressure = $MR/(V_c/T_c + V_k/T_k + V_r \ell n(T_h/T_k)/(T_h - T_k) + V_h/T_h + V_e/T_e),$

$Q_c = W_c = \oint p\, dV_c/d\theta\ \ d\theta$, $Q_e = W_e = \oint p\, dV_e/d\theta\ \ d\theta$,
$W = W_c + W_e$, η = efficiency = W/Q_e.

For sinusoidal volume variation this equation set can be explicitly integrated and closed-form solutions can be found, as carried out by Gustov Schmidt in 1871.

In a real machine the working spaces are more closely modeled by assuming adiabatic (no heat transfer), rather than isothermal, boundaries. The temperature changes produced by compression/expansion of the gas require that to maintain a constant temperature the working space would have to exchange heat with the surroundings perfectly. Since this is physically unrealistic, heat exchangers are necessary to exchange energy with the surroundings while the temperatures in the compression and expansion chambers are varying with pressure and volume. The adiabatic model is ideal in that the regenerators and heat exchangers are perfect (100% effective). The adiabatic model surpasses the isothermal model because the adiabatic condition more accurately describes actual machines, although the quantitative results are not significantly different. Non-isothermal behavior requires a much larger set of non-linear differential equations than does an isothermal. Closed form solutions are not possible, and computer-aided integration and iterative methods must be used.

The ideal adiabatic model improves on the ideal isothermal model by allowing the gas mass in the three heat exchangers to vary as changes in pressure occur, and also allows the temperatures in the compression and expansion chambers to vary with the pressure and volume.

To insure that cyclical steady state is achieved, the model is run over 10 cycles and only the results of the last cycle are reported.

The quasi-steady flow analysis goes into further detail on the heat exchangers by dividing the regenerator into two effective chambers with differing temperatures and pressures, and going into further detail on heat, mass, and energy transfer. The regenerators and heat exchangers, particularly the heater, are the most complicated components in a Stirling engine, and have the greatest effect on their performance. The temperature differential across the regenerator can be large, affecting the numerical stability of the analysis. Because of the numbers of equations in the analysis and their nonlinearity it is necessary to perform the analysis in an iterative process, and the integration must be performed over an number of cycles to insure that cyclic periodicity has been attained. It is recommended that the program first be run without the regenerator pressure drop, and then rerun with the regenerator pressure drop included, the no pressure drop results being used as new inputs for the various parameters.

The solution of the equation set requires numerically solving a set of differential equations. The program does this using the fourth-order Runge-Kutta method. This method is often used to solve initial-value problems. The adiabatic model, however, is a boundary-value problem. To overcome this apparent contradiction, arbitrary guesses are made for the initial conditions and several cycles are completed with the output from the previous cycle used as input to the subsequent cycle. In this manner, the correct solution is obtained when the input values equal the output values for a given cycle. This corresponds to cyclical steady state and usually it is achieved after 10 cycles.

The quasi-steady flow analysis goes further into detail by assuming realistic heat exchangers and regenerators while allowing the pressure drop and dissipation terms to be included. The analysis divides the regenerator into two effective chambers with differing temperatures and pressures, and goes into more detail on heat, mass, and energy transfer. The regenerators and heat exchangers, particularly the heater, are the most important components in a Stirling engine, and have the greatest effect on performance. The temperature differential across the regenerator can be large, affecting the numerical stability of the analysis. Because of the numbers of equations in the analysis and their nonlinearity it is necessary to perform the analysis in an iterative process, and the integration must be performed over an number of cycles to insure that cyclic periodicity has been attained. The equations are stiff, and the solution can become unstable. It is best to make gradual departures from a solution which is stable.

The analysis is separated into two parts: 1. the regenerator analysis and 2. the heat exchanger analysis. The regenerator analysis is the more simplified analysis because the gas within the heat exchangers remains at a constant temperature. The differential equations are solved in the same fashion as in the adiabatic model. The regenerator heat (the energy stored in the regenerator) should sum to zero over a complete cycle at steady state. To achieve this condition, the regenerator matrix temperatures are corrected through the use of a "matrix correction factor" entered by the user. This factor accelerates the convergence. The program's stability is tied to the matrix correction factor and improper choices can cause divergence. The pressure drop, and along with it the dissipation terms, can be included in the analysis at the user's discretion. These terms are of significant importance to the performance of a Stirling cycle. The regenerator matrix resembles a porous medium, consisting of a close-packed wire mesh. Compared to flow in a tube, a substantial pressure drop will result from this type of flow detrimentally affecting the performance. It is recommended the regenerator analysis be run first since it is somewhat simplified and then the more complicated heat exchanger analysis. This allows the heat exchanger analysis to use the output for the first initial guesses. Because of the highly increased complication to the system, the more accurate initial guesses (rather than the arbitrary first guess of all zeros) increase the stability of the solution. Overall, of the four possible choices the regenerator analysis without a pressure drop is the simplest, least realistic, and most stable analysis while the heat exchanger analysis with a pressure drop is the most complicated, most realistic, and least stable analysis.

Three different drive mechanisms can be selected. The sinusoidal drive motion corresponds to the use of a swashplate. The rhombic and yoke drives are described in Figures G-7 and G-8 of Appendix G. These figures should be consulted for a definition of the mechanism lengths.

The data to be input is the following:

Drive to be used for output files
Desired units
Desired application - Engine, heat pump-cooling, or heat pump-heating
Simulation model - Ideal isothermal, ideal adiabatic, or quasi-steady flow
Drive kinematics - Rhombic, sinusoidal, or Ross yoke
(The following is for a heat pump-cooling, quasi-steady flow, rhombic drive.)
Lengths of connecting rod, crank, eccentric; diameter of cylinder and rod.
Clearance volumes for compression and expansion spaces
Cooler configuration - Smooth pipes or smooth fins
Regenerator configuration - Tubular or annular
(The following is for a smooth pipe cooler and tubular regenerator)
Cooler external and internal diameters, length, number of tubes.
Regenerator porosity and wire diameter
Heater - smooth pipes or annulus
Gas - Hydrogen, helium, or air
Specify cooling/heating load or pressure
(The following is for the cooling load)
Thermal load
Temperatures of cooler and heater
Engine rpm
Simulation model - Regenerator, heat exchanger, or quit
(The following is for the regenerator model)
Matrix temperature convergence acceleration factor
Include the regenerator pressure drop in the analysis (Y/N)
Printing and convergence factors

Sample output for STIRL6.FOR, isothermal analysis - SI units
STIRLING HEAT PUMP - COOLING MODE (refrigeration)

```
Rhombic drive configuration
Lengths - Connecting rod, crank, eccentricity [mm]
    46.00   13.80   20.80
Diameters - cylinder, connecting rod [mm]
    69.90   9.52
Clearance volumes - compression, expansion
        28.68   30.52 [cm^3]
Swept volumes - compression, expansion
        114.13  120.82 [cm^3]
Advance angle              120.30 [degrees]
--------------------------------
Cooler - smooth pipes
Homogeneous bundle of pipes
Dinterior, length, number of pipes
  1.08    46.10 [mm]    312
Volume      13.18 [cm^3]
--------------------------------
Tubular regenerator
```

Tube Douter, Dinner, length, number of tubes
24.0 22.6 22.6 [mm] 8
Stacked wire mesh matrix
Porosity .697
Wire mesh diameter 40.0 [micrometers]
Hydraulic diameter .0920 [mm]
Total wetted area 21976.0 [cm^2]
Void volume 50.55 [cm^3]

Heater - smooth pipes
Homogeneous bundle of pipes
Dinterior, length, number of pipes
3.02 245.30 [mm] 40
Volume 70.28 [cm^3]

Gas is HELIUM
Desired Cooling requirement 8800.00 [W]
Engine speed 2503.00 [rpm]
Tcooler, Theater 35.50 22.00 [deg C]
Mean pressure 71.29 [bars]
Total mass of gas in engine 3.486 [gm]

Schmidt cycle analysis
Work -.01 [kJ] , Power -.39 [kW]
Qexp .21, Qcom -.22 [kJ] per cycle
Regenerator wall heat leakage -5.89 [W]
which is -.141E+00 [J] per cycle

Ideal isothermal cycle analysis
Work -10.05 [J] , Power -419.41 [W]
Qexp 228.27, Qcom -238.32 [J]
****

Mean pressure 65.88 [bars]
Total mass of gas in engine 3.221 [gm]

Schmidt cycle analysis
Work -.01 [kJ] , Power -.36 [kW]
Qexp .19, Qcom -.20 [kJ] per cycle
Regenerator wall heat leakage -5.89 [W]
which is -.141E+00 [J] per cycle

Ideal isothermal cycle analysis
Work -9.29 [J] , Power -387.59 [W]
Qexp 210.95, Qcom -220.24 [J]
Cooling COP 22.704 , EER 77.471

Sample output for STIRL6.FOR, adiabatic analysis - SI units
STIRLING HEAT PUMP - COOLING MODE (refrigeration)
Rhombic drive configuration
Lengths - Connecting rod, crank, eccentricity [mm]
46.00 13.80 20.80
Diameters - cylinder, connecting rod [mm]
69.90 9.52
Clearance volumes - compression, expansion

28.68 30.52 [cm^3]
Swept volumes - compression, expansion
114.13 120.82 [cm^3]
Advance angle 120.30 [degrees]

Cooler - smooth pipes
Homogeneous bundle of pipes
Dinterior, length, number of pipes
1.08 46.10 [mm] 312
Volume 13.18 [cm^3]

Tubular regenerator
Tube Douter, Dinner, length, number of tubes
24.0 22.6 22.6 [mm] 8
Stacked wire mesh matrix
Porosity .697
Wire mesh diameter 40.0 [micrometers]
Hydraulic diameter .0920 [mm]
Total wetted area 21976.0 [cm^2]
Void volume 50.55 [cm^3]

Heater - smooth pipes
Homogeneous bundle of pipes
Dinterior, length, number of pipes
3.02 245.30 [mm] 40
Volume 70.28 [cm^3]

Gas is HELIUM
Desired Cooling requirement 8800.00 [W]
Engine speed 2503.00 [rpm]
Tcooler, Theater 35.00 22.00 [deg C]
Mean pressure 71.29 [bars]
Total mass of gas in engine 3.486 [gm]

Schmidt cycle analysis
Work -.01 [kJ], Power -.39 [kW]
Qexp .21, Qcom -.22 [kJ] per cycle
Regenerator wall heat leakage -5.89 [W]
which is -.141E+00 [J] per cycle

Ideal adiabatic analysis

Cycle number 1
Power -2144.4 [W] , regenerator heat -.205E+01 [J]

Cycle number 2
Power -2421.4 [W] , regenerator heat -.813E+00 [J]

Cycle number 3
Power -2497.4 [W] , regenerator heat -.221E+00 [J]

Cycle number 4
Power -2516.7 [W] , regenerator heat -.479E-01 [J]

Cycle number 5
Power -2521.4 [W] , regenerator heat -.301E-02 [J]

Cycle number 6
Power -2522.5 [W] , regenerator heat .867E-02 [J]

Cycle number 7
Power -2522.8 [W] , regenerator heat .109E-01 [J]

Cycle number 8
Power -2522.8 [W] , regenerator heat .118E-01 [J]

Cycle number 9
Power -2522.9 [W] , regenerator heat .119E-01 [J]

Cycle number 10
Power -2522.9 [W] , regenerator heat .120E-01 [J]
****

Mean pressure 57.73 [bars]
Total mass of gas in engine 2.823 [gm]

Schmidt cycle analysis
Work -.01 [kJ] , Power -.31 [kW]
Qexp .17, Qcom -.18 [kJ] per cycle
Regenerator wall heat leakage -5.89 [W]
which is -.141E+00 [J] per cycle

Ideal adiabatic analysis

Cycle number 1
Power -1736.5 [W] , regenerator heat -.166E+01 [J]

Cycle number 2
Power -1960.9 [W] , regenerator heat -.658E+00 [J]

Cycle number 3
Power -2022.4 [W] , regenerator heat -.178E+00 [J]

Cycle number 4
Power -2038.0 [W] , regenerator heat -.386E-01 [J]

Cycle number 5
Power -2041.8 [W] , regenerator heat -.227E-02 [J]

Cycle number 6
Power -2042.7 [W] , regenerator heat .697E-02 [J]

Cycle number 7
Power -2042.9 [W] , regenerator heat .916E-02 [J]

Cycle number 8
Power -2042.9 [W] , regenerator heat .994E-02 [J]

Cycle number 9

Power -2043.0 [W] , regenerator heat .101E-01 [J]

Cycle number 10
Power -2043.0 [W] , regenerator heat .102E-01 [J]

Cooling COP 4.307 , EER 14.697

Sample output for STIRL6.FOR, quasi-steady flow analysis - SI units
STIRLING HEAT PUMP - COOLING MODE (refrigeration)

```
Rhombic drive configuration
Lengths - Connecting rod, crank, eccentricity [mm]
        46.00    13.80    20.80
Diameters - cylinder, connecting rod [mm]
       69.90     9.52
Clearance volumes - compression, expansion
                       28.68   30.52 [cm^3]
Swept volumes - compression, expansion
                 114.13  120.82 [cm^3]
Advance angle                120.30 [degrees]
```

```
Cooler - smooth pipes
Homogeneous bundle of pipes
Dinterior, length, number of pipes
  1.08     46.10 [mm]     312
Volume    13.18 [cm^3]
```

```
Tubular regenerator
Tube Douter, Dinner, length, number of tubes
     24.0    22.6    22.6 [mm]      8
Stacked wire mesh matrix
Porosity              .697
Wire mesh diameter       40.0 [micrometers]
Hydraulic diameter       .0920 [mm]
Total wetted area     21976.0 [cm^2]
Void volume              50.55 [cm^3]
```

```
Heater - smooth pipes
Homogeneous bundle of pipes
Dinterior, length, number of pipes
  3.02    245.30 [mm]      40
Volume    70.28 [cm^3]
```

```
Gas is HELIUM
Desired Cooling requirement      8800.00 [W]
Engine speed                     2503.00 [rpm]
Tcooler, Theater           35.00    22.00 [deg C]
Mean pressure                      71.29 [bars]
Total mass of gas in engine        3.486 [gm]
```

```
Schmidt cycle analysis
Work      -.01 [kJ] , Power      -.39 [kW]
Qexp       .21,      Qcom       -.22 [kJ]  per cycle
Regenerator wall heat leakage -5.89 [W]
        which is           -.141E+00 [J]  per cycle
```

Quasi-steady analysis
Matrix temperature convergence acceleration factor .1000
Regenerator analysis
Pressure drop included

Cycle number 1
Matrix temperatures 31.0 25.0 [deg C]
Regenerator heats .201E+02 .579E+01 [J]
Root mean square regenerator heat .2092E+02 [J]
Corrected matrix temperatures
Matrix temperatures 29.0 24.5 [deg C]
Power -2845.2 [W]

Cycle number 2
Matrix temperatures 29.0 24.2 [deg C]
Regenerator heats .108E+01 .631E+01 [J]
Root mean square regenerator heat .6397E+01 [J]
Corrected matrix temperatures
Matrix temperatures 28.9 23.6 [deg C]
Power -3113.0 [W]

Cycle number 3
Matrix temperatures 28.7 23.6 [deg C]
Regenerator heats .355E+01 .330E+00 [J]
Root mean square regenerator heat .3570E+01 [J]
Corrected matrix temperatures
Matrix temperatures 28.4 23.5 [deg C]
Power -3189.4 [W]

Cycle number 4
Matrix temperatures 28.4 23.5 [deg C]
Regenerator heats .228E+00 .154E+01 [J]
Root mean square regenerator heat .1556E+01 [J]
Corrected matrix temperatures
Matrix temperatures 28.3 23.3 [deg C]
Power -3209.2 [W]

Cycle number 5
Matrix temperatures 28.3 23.3 [deg C]
Regenerator heats .715E+00 -.566E-01 [J]
Root mean square regenerator heat .7170E+00 [J]
Corrected matrix temperatures
Matrix temperatures 28.2 23.3 [deg C]
Power -3213.4 [W]

Cycle number 6
Matrix temperatures 28.2 23.3 [deg C]
Regenerator heats .152E-01 .343E+00 [J]
Root mean square regenerator heat .3436E+00 [J]
Corrected matrix temperatures
Matrix temperatures 28.2 23.3 [deg C]
Power -3214.6 [W]

Cycle number 7
Matrix temperatures 28.2 23.3 [deg C]
Regenerator heats .152E+00 -.347E-01 [J]
Root mean square regenerator heat .1557E+00 [J]
Corrected matrix temperatures
Matrix temperatures 28.2 23.3 [deg C]
Power -3214.6 [W]

Cycle number 8
Matrix temperatures 28.2 23.3 [deg C]
Regenerator heats -.794E-02 .739E-01 [J]
Root mean square regenerator heat .7436E-01 [J]
Corrected matrix temperatures
Matrix temperatures 28.2 23.3 [deg C]
Power -3214.7 [W]

Cycle number 9
Matrix temperatures 28.2 23.3 [deg C]
Regenerator heats .334E-01 -.121E-01 [J]
Root mean square regenerator heat .3557E-01 [J]
Corrected matrix temperatures
Matrix temperatures 28.2 23.3 [deg C]
Power -3214.7 [W]

Cycle number 10
Matrix temperatures 28.2 23.3 [deg C]
Regenerator heats -.630E-02 .165E-01 [J]
Root mean square regenerator heat .1762E-01 [J]
Corrected matrix temperatures
Matrix temperatures 28.2 23.3 [deg C]
Power -3214.7 [W]
****

Mean pressure 70.32 [bars]
Total mass of gas in engine 3.438 [gm]

Schmidt cycle analysis
Work -.01 [kJ] , Power -.38 [kW]
Qexp .21, Qcom -.22 [kJ] per cycle
Regenerator wall heat leakage -5.89 [W]
which is -.141E+00 [J] per cycle

Quasi-steady analysis

Cycle number 1
Matrix temperatures 31.0 25.0 [deg C]
Regenerator heats .198E+02 .574E+01 [J]
Root mean square regenerator heat .2058E+02 [J]
Corrected matrix temperatures
Matrix temperatures 29.0 24.5 [deg C]
Power -2810.1 [W]

Cycle number 2
Matrix temperatures 29.0 24.2 [deg C]
Regenerator heats .129E+01 .625E+01 [J]

Root mean square regenerator heat .6381E+01 [J]
Corrected matrix temperatures
Matrix temperatures 28.9 23.6 [deg C]
Power -3074.3 [W]

Cycle number 3
Matrix temperatures 28.7 23.6 [deg C]
Regenerator heats .349E+01 .455E+00 [J]
Root mean square regenerator heat .3516E+01 [J]
Corrected matrix temperatures
Matrix temperatures 28.4 23.5 [deg C]
Power -3149.6 [W]

Cycle number 4
Matrix temperatures 28.4 23.5 [deg C]
Regenerator heats .312E+00 .150E+01 [J]
Root mean square regenerator heat .1532E+01 [J]
Corrected matrix temperatures
Matrix temperatures 28.3 23.3 [deg C]
Power -3169.1 [W]

Cycle number 5
Matrix temperatures 28.3 23.3 [deg C]
Regenerator heats .694E+00 .349E-02 [J]
Root mean square regenerator heat .6940E+00 [J]
Corrected matrix temperatures
Matrix temperatures 28.3 23.3 [deg C]
Power -3173.3 [W]

Cycle number 6
Matrix temperatures 28.3 23.3 [deg C]
Regenerator heats .378E-01 .325E+00 [J]
Root mean square regenerator heat .3270E+00 [J]
Corrected matrix temperatures
Matrix temperatures 28.2 23.3 [deg C]
Power -3174.3 [W]

Cycle number 7
Matrix temperatures 28.2 23.3 [deg C]
Regenerator heats .148E+00 -.194E-01 [J]
Root mean square regenerator heat .1495E+00 [J]
Corrected matrix temperatures
Matrix temperatures 28.2 23.3 [deg C]
Power -3174.4 [W]

Cycle number 8
Matrix temperatures 28.2 23.3 [deg C]
Regenerator heats -.950E-04 .708E-01 [J]
Root mean square regenerator heat .7080E-01 [J]
Corrected matrix temperatures
Matrix temperatures 28.2 23.3 [deg C]
Power -3174.5 [W]

Cycle number 9

Matrix temperatures 28.2 23.3 [deg C]
Regenerator heats .304E-01 -.884E-02 [J]
Root mean square regenerator heat .3163E-01 [J]
Corrected matrix temperatures
Matrix temperatures 28.2 23.3 [deg C]
Power -3174.5 [W]

Cycle number 10
Matrix temperatures 28.2 23.3 [deg C]
Regenerator heats -.135E-02 .116E-01 [J]
Root mean square regenerator heat .1167E-01 [J]
Corrected matrix temperatures
Matrix temperatures 28.2 23.3 [deg C]
Power -3174.5 [W]

Cooling COP 2.772, EER 9.458

Sample output for STIRL6.FOR, isothermal analysis - English units

STIRLING HEAT PUMP - COOLING MODE (refrigeration)

Rhombic drive configuration
Lengths - Connecting rod, crank, eccentricity [in]
1.81 .54 .82
Diameters - cylinder, connecting rod [in]
2.75 .37
Clearance volumes - compression, expansion
1.75 1.86 [in^3]
Swept volumes - compression, expansion
6.93 7.32 [in^3]
Advance angle 120.32 [degrees]

Cooler - smooth pipes
Homogeneous bundle of pipes
Dinterior, length, number of pipes
.04 1.82 [in] 312
Volume .80 [in^3]

Tubular regenerator
Tube Douter, Dinner, length, number of tubes
.9 .9 .9 [in] 8
Stacked wire mesh matrix
Porosity .697
Wire mesh diameter 1575.0 [microinches]
Hydraulic diameter .0036 [in]
Total wetted area 3408.6 [in^2]
Void volume 3.09 [in^3]

Heater - smooth pipes
Homogeneous bundle of pipes
Dinterior, length, number of pipes
.12 9.66 [in] 40
Volume 4.29 [in^3]

Gas is HELIUM

Desired Cooling requirement 30000.00 [Btu/hr]
Engine speed 2503.00 [rpm]
Tcooler, Theater 95.00 71.60 [deg F]
Mean pressure 1043.98 [psia]
Total mass of gas in engine .008 [lbm]

Schmidt cycle analysis
Work -.01 [Btu], Power -1321.36 [Btu/hr]
Qexp .20, Qcom -.21 [Btu] per cycle
Regenerator wall heat leakage -20.07 [Btu/hr]
which is -.134E-03 [Btu] per cycle

Ideal isothermal cycle analysis
Work -.01 [Btu], Power -1428.64 [Btu/hr]
Qexp .22, Qcom -.23 [Btu]
****

Mean pressure 965.58 [psia]
Total mass of gas in engine .007 [lbm]

Schmidt cycle analysis
Work -.01 [Btu], Power -1222.13 [Btu/hr]
Qexp .18, Qcom -.19 [Btu] per cycle
Regenerator wall heat leakage -20.07 [Btu/hr]
which is -.134E-03 [Btu] per cycle

Ideal isothermal cycle analysis
Work -.01 [Btu], Power -1321.32 [Btu/hr]
Qexp .20, Qcom -.21 [Btu]
Cooling COP 22.705 , EER 77.471

Sample output for STIRL6.FOR, adiabatic analysis - English units
STIRLING HEAT PUMP - COOLING MODE (refrigeration)
Rhombic drive configuration
Lengths - Connecting rod, crank, eccentricity [in]
1.81 .54 .82
Diameters - cylinder, connecting rod [in]
2.75 .37
Clearance volumes - compression, expansion
1.75 1.86 [in^3]
Swept volumes - compression, expansion
6.93 7.32 [in^3]
Advance angle 120.32 [degrees]

Cooler - smooth pipes
Homogeneous bundle of pipes
Dinterior, length, number of pipes
.04 1.82 [in] 312
Volume .80 [in^3]

Tubular regenerator
Tube Douter, Dinner, length, number of tubes
.9 .9 .9 [in] 8
Stacked wire mesh matrix

Porosity .697
Wire mesh diameter 1575.0 [microinches]
Hydraulic diameter .0036 [in]
Total wetted area 3408.6 [in^2]
Void volume 3.09 [in^3]

Heater - smooth pipes
Homogeneous bundle of pipes
Dinterior, length, number of pipes
.12 9.66 [in] 40
Volume 4.29 [in^3]

Gas is HELIUM
Desired Cooling requirement 30000.00 [Btu/hr]
Engine speed 2503.00 [rpm]
Tcooler, Theater 95.00 71.60 [deg F]
Mean pressure 1043.98 [psia]
Total mass of gas in engine .008 [lbm]

Schmidt cycle analysis
Work -.01 [Btu], Power -1321.36 [Btu/hr]
Qexp .20, Qcom -.21 [Btu] per cycle
Regenerator wall heat leakage -20.07 [Btu/hr]
which is -.134E-03 [Btu] per cycle

Ideal adiabatic analysis

Cycle number 1
Power -7274.1 [Btu/hr], regenerator heat -.191E-02 [Btu]

Cycle number 2
Power -8209.1 [Btu/hr], regenerator heat -.769E-03 [Btu]

Cycle number 3
Power -8466.5 [Btu/hr], regenerator heat -.211E-03 [Btu]

Cycle number 4
Power -8532.2 [Btu/hr], regenerator heat -.461E-04 [Btu]

Cycle number 5
Power -8548.3 [Btu/hr], regenerator heat -.280E-05 [Btu]

Cycle number 6
Power -8552.2 [Btu/hr], regenerator heat .807E-05 [Btu]

Cycle number 7
Power -8553.2 [Btu/hr], regenerator heat .104E-04 [Btu]

Cycle number 8
Power -8553.3 [Btu/hr], regenerator heat .112E-04 [Btu]

Cycle number 9
Power -8553.4 [Btu/hr], regenerator heat .115E-04 [Btu]

Cycle number 10
Power -8553.4 [Btu/hr], regenerator heat .115E-04 [Btu]
****

Mean pressure 845.83 [psia]
Total mass of gas in engine .006 [lbm]

Schmidt cycle analysis
Work -.01 [Btu], Power -1070.57 [Btu/hr]
Qexp .16, Qcom -.17 [Btu] per cycle
Regenerator wall heat leakage -20.07 [Btu/hr]
which is -.134E-03 [Btu] per cycle

Ideal adiabatic analysis

Cycle number 1
Power -5893.5 [Btu/hr], regenerator heat -.155E-02 [Btu]

Cycle number 2
Power -6650.9 [Btu/hr], regenerator heat -.622E-03 [Btu]

Cycle number 3
Power -6859.6 [Btu/hr], regenerator heat -.170E-03 [Btu]

Cycle number 4
Power -6912.7 [Btu/hr], regenerator heat -.377E-04 [Btu]

Cycle number 5
Power -6925.8 [Btu/hr], regenerator heat -.212E-05 [Btu]

Cycle number 6
Power -6929.0 [Btu/hr], regenerator heat .634E-05 [Btu]

Cycle number 7
Power -6929.8 [Btu/hr], regenerator heat .850E-05 [Btu]

Cycle number 8
Power -6930.0 [Btu/hr], regenerator heat .909E-05 [Btu]

Cycle number 9
Power -6930.0 [Btu/hr], regenerator heat .914E-05 [Btu]

Cycle number 10
Power -6930.0 [Btu/hr], regenerator heat .913E-05 [Btu]

Cooling COP 4.329 , EER 14.771

Sample output for STIRL6.FOR, quasi-steady analysis - English units
STIRLING HEAT PUMP - COOLING MODE (refrigeration)
Rhombic drive configuration
Lengths - Connecting rod, crank, eccentricity [in]
1.81 .54 .82
Diameters - cylinder, connecting rod [in]
2.75 .37

Clearance volumes - compression, expansion
1.75 1.86 [in^3]
Swept volumes - compression, expansion
6.93 7.32 [in^3]
Advance angle 120.32 [degrees]

Cooler - smooth pipes
Homogeneous bundle of pipes
Dinterior, length, number of pipes
.04 1.82 [in] 312
Volume .80 [in^3]

Tubular regenerator
Tube Douter, Dinner, length, number of tubes
.9 .9 .9 [in] 8
Stacked wire mesh matrix
Porosity .697
Wire mesh diameter 1575.0 [microinches]
Hydraulic diameter .0036 [in]
Total wetted area 3408.6 [in^2]
Void volume 3.09 [in^3]

Heater - smooth pipes
Homogeneous bundle of pipes
Dinterior, length, number of pipes
.12 9.66 [in] 40
Volume 4.29 [in^3]

Gas is HELIUM
Desired Cooling requirement 30000.00 [Btu/hr]
Engine speed 2503.00 [rpm]
Tcooler, Theater 95.00 71.60 [deg F]
Mean pressure 1043.98 [psia]
Total mass of gas in engine .008 [lbm]

Schmidt cycle analysis
Work -.01 [Btu], Power -1321.36 [Btu/hr]
Qexp .20, Qcom -.21 [Btu] per cycle
Regenerator wall heat leakage -20.07 [Btu/hr]
which is -.134E-03 [Btu] per cycle

Quasi-steady analysis
Matrix temperature convergence acceleration factor .1000
Regenerator analysis
Pressure drop included

Cycle number 1
Matrix temperatures 87.8 77.1 [deg F]
Regenerator heats .190E-01 .543E-02 [Btu]
Root mean square regenerator heat .1980E-01 [Btu]
Corrected matrix temperatures
Matrix temperatures 84.2 76.0 [deg F]
Power -9637.1 [Btu/hr]

Cycle number 2
Matrix temperatures 84.1 75.6 [deg F]
Regenerator heats .930E-03 .594E-02 [Btu]
Root mean square regenerator heat .6014E-02 [Btu]
Corrected matrix temperatures
Matrix temperatures 84.0 74.5 [deg F]
Power -10540.9 [Btu/hr]

Cycle number 3
Matrix temperatures 83.7 74.5 [deg F]
Regenerator heats .337E-02 .271E-03 [Btu]
Root mean square regenerator heat .3384E-02 [Btu]
Corrected matrix temperatures
Matrix temperatures 83.1 74.4 [deg F]
Power -10799.9 [Btu/hr]

Cycle number 4
Matrix temperatures 83.1 74.3 [deg F]
Regenerator heats .187E-03 .147E-02 [Btu]
Root mean square regenerator heat .1483E-02 [Btu]
Corrected matrix temperatures
Matrix temperatures 83.0 74.0 [deg F]
Power -10867.5 [Btu/hr]

Cycle number 5
Matrix temperatures 83.0 74.0 [deg F]
Regenerator heats .682E-03 -.724E-04 [Btu]
Root mean square regenerator heat .6860E-03 [Btu]
Corrected matrix temperatures
Matrix temperatures 82.9 74.1 [deg F]
Power -10881.9 [Btu/hr]

Cycle number 6
Matrix temperatures 82.9 74.0 [deg F]
Regenerator heats .310E-05 .330E-03 [Btu]
Root mean square regenerator heat .3299E-03 [Btu]
Corrected matrix temperatures
Matrix temperatures 82.9 74.0 [deg F]
Power -10886.0 [Btu/hr]

Cycle number 7
Matrix temperatures 82.8 74.0 [deg F]
Regenerator heats .138E-03 -.414E-04 [Btu]
Root mean square regenerator heat .1445E-03 [Btu]
Corrected matrix temperatures
Matrix temperatures 82.8 74.0 [deg F]
Power -10886.3 [Btu/hr]

Cycle number 8
Matrix temperatures 82.8 74.0 [deg F]
Regenerator heats -.128E-04 .745E-04 [Btu]
Root mean square regenerator heat .7560E-04 [Btu]
Corrected matrix temperatures
Matrix temperatures 82.8 74.0 [deg F]

Power -10886.8 [Btu/hr]

Cycle number 9
Matrix temperatures 82.8 74.0 [deg F]
Regenerator heats .309E-04 -.143E-04 [Btu]
Root mean square regenerator heat .3403E-04 [Btu]
Corrected matrix temperatures
Matrix temperatures 82.8 74.0 [deg F]
Power -10886.7 [Btu/hr]

Cycle number 10
Matrix temperatures 82.8 74.0 [deg F]
Regenerator heats -.211E-05 .159E-04 [Btu]
Root mean square regenerator heat .1602E-04 [Btu]
Corrected matrix temperatures
Matrix temperatures 82.8 74.0 [deg F]
Power -10886.7 [Btu/hr]
****

Mean pressure 1029.04 [psia]
Total mass of gas in engine .008 [lbm]

Schmidt cycle analysis
Work -.01 [Btu], Power -1302.45 [Btu/hr]
Qexp .20, Qcom -.21 [Btu] per cycle
Regenerator wall heat leakage -20.07 [Btu/hr]
which is -.134E-03 [Btu] per cycle

Quasi-steady analysis

Cycle number 1
Matrix temperatures 87.8 77.1 [deg F]
Regenerator heats .187E-01 .538E-02 [Btu]
Root mean square regenerator heat .1947E-01 [Btu]
Corrected matrix temperatures
Matrix temperatures 84.3 76.1 [deg F]
Power -9512.6 [Btu/hr]

Cycle number 2
Matrix temperatures 84.2 75.6 [deg F]
Regenerator heats .113E-02 .589E-02 [Btu]
Root mean square regenerator heat .6002E-02 [Btu]
Corrected matrix temperatures
Matrix temperatures 84.0 74.5 [deg F]
Power -10403.7 [Btu/hr]

Cycle number 3
Matrix temperatures 83.8 74.5 [deg F]
Regenerator heats .330E-02 .392E-03 [Btu]
Root mean square regenerator heat .3327E-02 [Btu]
Corrected matrix temperatures
Matrix temperatures 83.1 74.4 [deg F]
Power -10659.2 [Btu/hr]

Cycle number 4

```
Matrix temperatures            83.1      74.3 [deg F]
Regenerator heats           .272E-03   .143E-02 [Btu]
Root mean square regenerator heat       .1452E-02 [Btu]
 Corrected matrix temperatures
Matrix temperatures            83.1      74.1 [deg F]
Power                       -10725.7 [Btu/hr]
```

```
Cycle number    5
Matrix temperatures            83.0      74.1 [deg F]
Regenerator heats           .659E-03  -.113E-04 [Btu]
Root mean square regenerator heat       .6595E-03 [Btu]
 Corrected matrix temperatures
Matrix temperatures            82.9      74.1 [deg F]
Power                       -10739.9 [Btu/hr]
```

```
Cycle number    6
Matrix temperatures            82.9      74.0 [deg F]
Regenerator heats           .365E-04   .311E-03 [Btu]
Root mean square regenerator heat       .3133E-03 [Btu]
 Corrected matrix temperatures
Matrix temperatures            82.9      74.0 [deg F]
Power                       -10743.7 [Btu/hr]
```

```
Cycle number    7
Matrix temperatures            82.9      74.0 [deg F]
Regenerator heats           .136E-03  -.215E-04 [Btu]
Root mean square regenerator heat       .1381E-03 [Btu]
 Corrected matrix temperatures
Matrix temperatures            82.8      74.0 [deg F]
Power                       -10744.0 [Btu/hr]
```

```
Cycle number    8
Matrix temperatures            82.9      74.0 [deg F]
Regenerator heats           .326E-05   .666E-04 [Btu]
Root mean square regenerator heat       .6666E-04 [Btu]
 Corrected matrix temperatures
Matrix temperatures            82.8      74.0 [deg F]
Power                       -10744.2 [Btu/hr]
```

```
Cycle number    9
Matrix temperatures            82.8      74.0 [deg F]
Regenerator heats           .283E-04  -.991E-05 [Btu]
oot mean square regenerator heat       .2995E-04 [Btu]
 Corrected matrix temperatures
Matrix temperatures            82.8      74.0 [deg F]
Power                       -10744.0 [Btu/hr]
```

```
Cycle number    10
Matrix temperatures            82.8      74.0 [deg F]
Regenerator heats          -.174E-05   .144E-04 [Btu]
Root mean square regenerator heat       .1447E-04 [Btu]
 Corrected matrix temperatures
Matrix temperatures            82.8      74.0 [deg F]
Power                       -10744.1 [Btu/hr]
```

Cooling COP 2.792, EER 9.527

K.4 Thermoelectric Heat Pump Program - TEEFFECT.FOR

This program provides the operator with three options:

1. List the electrical and thermal properties of modules produced by two manufacturers of thermoelectric modules, Melcor and ITI-Ferrotec.
2. List the figure of merit (Z and ZT) of the modules in option 1.
3. Design a thermoelectric heat pump.

Option 1 allows the operator to compute the electrical resistance, thermal conductance, and Seebeck coefficient of a given module. The module can be either a module listed in the catalogs of Melcor or ITI-Ferrotec, or the operator can respond with individual values of these coefficients.

Option 2 presents figures of merit of the chosen module as a function of temperature.

The third option allows the operator to select a particular thermoelectric module, cold and hot operating temperatures, and the thermal resistances of the heat exchangers on either side of the module. For a variety of electrical currents the hot and cold side heat flows are calculated along with the coefficients of performance and the voltage. Specifying either the heating or cooling load allows calculation of the number of thermoelectric modules needed for that load.

The data to be input is the following:

Drive to be used for output files
Desired units
Options - Design of single or multiple-stage heat pumps, properties of a single stage module, performance coefficient of a single-stage module
Module manufacturer - ITI Ferrotec, Melcor, custom
(The following is for a Melcor module)
Number of thermocouples per module
Geometric factor (ratio of thermocouple area to length of legs)
Cold and hot temperatures
Cold and hot thermal resistances
Cooling and heating loads
Maximum current to be investigated
Current step size to be used

The calculated coefficients of performance are realistic, providing that the thermal resistances of the heat exchangers are properly included and that care is taken in the manufacturing process to ensure that good thermal contact is made and maintained between the heat exchangers and thermoelectric modules.

Sample output for TEEFFECT.FOR - SI units

THERMO-ELECTRIC HEAT PUMP ANALYSIS

Module manufacturer is MELCOR.
The number of thermocouples per module is 31
The geometric factor for your module is 1.2550000

HEAT PUMP DESIGN - SINGLE STAGE MODULES

Tambient cold = 22.00, Tambient hot = 35.00 [deg C]
Rcold = .0000, Rhot = .0000 [deg C/W]
COPhot = COPcold + 1, EERhot = EERcold + 3.412
Desired cooling load is 20. [kW]
Desired heating load is 10. [kW]

Current	Module Temp.		Q per module		COP	EER	Voltage	Number of	
[amps]	[deg C]		[W]				[volts]	modules	
cool	heat	cool	heat	cool	cool		cool	heat	
5.0	22.00	35.00	2.17	4.40	.970	3.309	.447	9228	2272
10.0	22.00	35.00	19.57	26.80	2.709	9.243	.723	1022	374
15.0	22.00	35.00	35.60	50.57	2.378	8.114	.998	562	198
20.0	22.00	35.00	50.25	75.72	1.973	6.732	1.274	399	133
25.0	22.00	35.00	63.52	102.25	1.640	5.597	1.549	315	98
30.0	22.00	35.00	75.42	130.15	1.378	4.701	1.825	266	77
35.0	22.00	35.00	85.94	159.44	1.169	3.989	2.100	233	63
40.0	22.00	35.00	95.08	190.10	1.001	3.414	2.376	211	53
45.0	22.00	35.00	102.84	222.14	.862	2.941	2.651	195	46
50.0	22.00	35.00	109.22	255.55	.746	2.547	2.927	184	40
55.0	22.00	35.00	114.23	290.35	.649	2.213	3.202	176	35
60.0	22.00	35.00	117.86	326.52	.565	1.927	3.478	170	31
65.0	22.00	35.00	120.11	364.06	.492	1.680	3.753	167	28
70.0	22.00	35.00	120.99	402.99	.429	1.464	4.029	166	25
75.0	22.00	35.00	120.48	443.29	.373	1.274	4.304	166	23

Module manufacturer is MELCOR.
The number of thermocouples per module is 31
The geometric factor for your module is 1.2550000

HEAT PUMP DESIGN - SINGLE STAGE MODULES

Tambient cold = 22.00, Tambient hot = 35.00 [deg C]
Rcold = .1000, Rhot = .1000 [deg C/W]
COPhot = COPcold + 1, EERhot = EERcold + 3.412
Desired cooling load is 20. [kW]
Desired heating load is 10. [kW]

Current	Module Temp.		Q per module		COP	EER	Voltage	Number of	
[amps]	[deg C]		[W]				[volts]	modules	
	cool	heat	cool	heat	cool	cool		cool	heat
5.0	21.85	35.38	1.49	3.76	.657	2.243	.454	13402	2658
10.0	20.53	37.24	14.67	22.40	1.899	6.481	.773	1364	447
15.0	19.37	39.27	26.34	42.73	1.607	5.483	1.093	760	235
20.0	18.35	41.48	36.50	64.81	1.290	4.401	1.415	548	155
25.0	17.48	43.87	45.18	88.72	1.038	3.543	1.740	443	113
30.0	16.76	46.45	52.38	114.52	.844	2.878	2.070	382	88

35.0	16.19	49.23	58.10	142.35	.690	2.353	2.407	345	71
40.0	15.77	52.23	62.32	172.25	.567	1.934	2.748	321	59
45.0	15.50	55.44	65.02	204.38	.467	1.592	3.097	308	49
50.0	15.38	58.88	66.18	238.85	.383	1.308	3.453	303	42
55.0	15.42	62.58	65.76	275.80	.313	1.068	3.819	305	37
60.0	15.63	66.54	63.70	315.41	.253	.864	4.194	314	32
65.0	16.01	70.79	59.95	357.86	.201	.687	4.582	334	28
70.0	16.56	75.33	54.42	403.33	.156	.532	4.983	368	25
75.0	17.30	80.23	46.97	452.27	.116	.395	5.404	426	23

Module manufacturer is MELCOR.
The geometric factor for your module is 1.2550000

HEAT PUMP DESIGN - MULTIPLE STAGE MODULES
You are using modules with 3 stages.
The number of thermocouples in stage 1 is 127
The number of thermocouples in stage 2 is 71
The number of thermocouples in stage 3 is 31

Tambient cold = 22.00, Tambient hot = 35.00 [deg C]
Rcold = .1000, Rhot = .1000 [deg C/W]
COPhot = COPcold + 1, EERhot = EERcold + 3.412
Desired cooling load is 20. [kW]
Desired heating load is 10. [kW]

Current [amps]	Module Temp. [deg C] cool	heat	Q per module [W] cool	heat	COP cool	EER cool	Voltage [volts]	Number of modules cool	heat
1.5	21.97	35.18	.27	1.84	.168	.574	1.052	75306	5423
2.0	21.69	35.57	3.07	5.70	1.172	4.000	1.311	6508	1756
2.5	21.42	35.97	5.81	9.73	1.483	5.060	1.568	3441	1028
3.0	21.15	36.40	8.49	13.96	1.551	5.292	1.824	2357	717
3.5	20.89	36.84	11.09	18.37	1.525	5.203	2.079	1803	545
4.0	20.64	37.30	13.64	22.97	1.462	4.987	2.333	1467	436
4.5	20.39	37.78	16.12	27.76	1.385	4.727	2.586	1241	361
5.0	20.15	38.27	18.54	32.73	1.306	4.457	2.838	1079	306

Sample output for TEEFFECT.FOR - English units

THERMO-ELECTRIC HEAT PUMP ANALYSIS

Module manufacturer is MELCOR.
The number of thermocouples per module is 31
The geometric factor for your module is 1.2550000

HEAT PUMP DESIGN - SINGLE STAGE MODULES

Tambient cold = 71.00, Tambient hot = 95.00 [deg F]
Rcold = .0000, Rhot = .0000 [deg F-hr/Btu]
COPhot = COPcold + 1, EERhot = EERcold + 3.412
Desired cooling load is 68250. [Btu/hr]
Desired heating load is 34125. [Btu/hr]

Current [amps] cool	Module Temp. [deg F] heat	cool	Q per module [Btu/hr] heat	cool	COP cool	EER	Voltage [volts] cool	Number of modules heat	
5.0	71.00	95.00	5.83	13.53	.758	2.586	.451	11698	2522

10.0	71.00	95.00	65.14	89.92	2.628	8.967	.726	1048	380
15.0	71.00	95.00	119.74	171.01	2.335	7.969	1.002	570	200
20.0	71.00	95.00	169.65	256.80	1.947	6.643	1.277	403	133
25.0	71.00	95.00	214.86	347.28	1.623	5.537	1.552	318	99
30.0	71.00	95.00	255.38	442.46	1.365	4.658	1.828	268	78
35.0	71.00	95.00	291.20	542.33	1.160	3.957	2.103	235	63
40.0	71.00	95.00	322.32	646.91	.993	3.388	2.378	212	53
45.0	71.00	95.00	348.75	756.17	.856	2.921	2.653	196	46
50.0	71.00	95.00	370.48	870.14	.741	2.530	2.929	185	40
55.0	71.00	95.00	387.51	988.80	.644	2.199	3.204	177	35
60.0	71.00	95.00	399.85	1112.16	.561	1.915	3.479	171	31
65.0	71.00	95.00	407.49	1240.21	.489	1.670	3.755	168	28
70.0	71.00	95.00	410.43	1372.96	.426	1.455	4.030	167	25
75.0	71.00	95.00	408.68	1510.41	.371	1.266	4.305	168	23

Module manufacturer is MELCOR.
The number of thermocouples per module is 31
The geometric factor for your module is 1.2550000

HEAT PUMP DESIGN - SINGLE STAGE MODULES
Tambient cold = 22.00, Tambient hot = 35.00 [deg C]
Rcold = .1000, Rhot = .1000 [deg C/W]
COPhot = COPcold + 1, EERhot = EERcold + 3.412
Desired cooling load is 20. [kW]
Desired heating load is 10. [kW]

Current	Module Temp.		Q per module		COP	EER	Voltage	Number of	
[amps]	[deg C]		[W]				[volts]	modules	
cool	heat	cool	heat	cool	cool		cool	heat	
5.0	21.85	35.38	1.49	3.76	.657	2.243	.454	13402	2658
10.0	20.53	37.24	4.67	22.40	1.899	6.481	.773	1364	447
15.0	19.37	39.27	26.34	42.73	1.607	5.483	1.093	760	235
20.0	18.35	41.48	36.50	64.81	1.290	4.401	1.415	548	155
25.0	17.48	43.87	45.18	88.72	1.038	3.543	1.740	443	113
30.0	16.76	46.45	52.38	114.52	.844	2.878	2.070	382	88
35.0	16.19	49.23	58.10	142.35	.690	2.353	2.407	345	71
40.0	15.77	52.23	62.32	172.25	.567	1.934	2.748	321	59
45.0	15.50	55.44	65.02	204.38	.467	1.592	3.097	308	49
50.0	15.38	58.88	66.18	238.85	.383	1.308	3.453	303	42
55.0	15.42	62.58	65.76	275.80	.313	1.068	3.819	305	37
60.0	15.63	66.54	63.70	315.41	.253	.864	4.194	314	32
65.0	16.01	70.79	59.95	357.86	.201	.687	4.582	334	28
70.0	16.56	75.33	54.42	403.33	.156	.532	4.983	368	25
75.0	17.30	80.23	46.97	452.27	.116	.395	5.404	426	23

THERMO-ELECTRIC HEAT PUMP ANALYSIS
Module manufacturer is MELCOR.
The geometric factor for your module is 1.2550000
HEAT PUMP DESIGN - MULTIPLE STAGE MODULES
You are using modules with 3 stages.
The number of thermocouples in stage 1 is 127
The number of thermocouples in stage 2 is 71
The number of thermocouples in stage 3 is 31

Tambient cold = 22.00, Tambient hot = 46.00 [deg C]
Rcold = .1000, Rhot = .1000 [deg C/W]
COPhot = COPcold + 1, EERhot = EERcold + 3.412
Desired cooling load is 20. [kW]
Desired heating load is 10. [kW]

Current [amps]	Module Temp. [deg C]		Q per module [W]		COP	EER	Voltage [volts]	Number of modules	
cool	heat	cool	heat	cool	cool			cool	heat
3.0	21.83	46.81	1.68	8.06	.264	.899	2.126	11899	1241
3.5	21.57	47.27	4.34	12.71	.518	1.768	2.392	4611	787
4.0	21.31	47.76	6.93	17.56	.652	2.225	2.657	2887	570
4.5	21.05	48.26	9.46	22.60	.719	2.454	2.921	2116	443
5.0	20.81	48.78	11.92	27.84	.748	2.554	3.185	1679	360

THERMO-ELECTRIC HEAT PUMP ANALYSIS
Module manufacturer is MELCOR.
The number of thermocouples per module is 31
The geometric factor for your module is 1.255000
HEAT PUMP DESIGN - SINGLE STAGE MODULES
Tambient cold = 71.60, Tambient hot = 95.00 [deg F]
Rcold = .0527, Rhot = .0527 [deg F-hr/Btu]
COPhot = COPcold + 1, EERhot = EERcold + 3.412
Desired cooling load is 68246. [Btu/hr]
Desired heating load is 34123. [Btu/hr]
Temperature [° F]

Current [amps]	Module Temp. [deg F]		Q per module [Btu/hr]		COP	EER	Voltage [volts]	Number of modules	
cool	heat	cool	heat	cool	cool			cool	heat
5.0	71.33	95.68	5.09	12.84	.657	2.243	.454	13399	2658
10.0	68.96	99.03	50.08	76.44	1.900	6.483	.772	1363	447
15.0	66.86	102.68	89.89	145.82	1.608	5.485	1.092	760	235
20.0	65.03	106.66	124.59	221.17	1.290	4.403	1.415	548	155
25.0	63.47	110.95	154.22	302.75	1.039	3.545	1.740	443	113
30.0	62.18	115.60	178.79	390.80	.844	2.880	2.069	382	88
35.0	61.15	120.60	198.32	485.74	.690	2.354	2.407	345	71
40.0	60.39	125.98	212.73	587.80	.567	1.935	2.748	321	59
45.0	59.90	131.75	221.98	697.40	.467	1.593	3.096	308	49
50.0	59.69	137.95	225.95	815.01	.384	1.309	3.452	303	42
55.0	59.77	144.60	224.52	941.10	.313	1.069	3.818	304	37
60.0	60.14	151.72	217.52	1076.23	.253	.864	4.194	314	32
65.0	60.81	159.35	204.73	1221.03	.201	.688	4.581	334	28
70.0	61.80	167.52	185.89	1376.18	.156	.533	4.982	368	25
75.0	63.14	176.32	160.48	1543.11	.116	.396	5.403	426	23

THERMO-ELECTRIC HEAT PUMP ANALYSIS
Module manufacturer is MELCOR.
The geometric factor for your module is 1.255000
HEAT PUMP DESIGN - MULTIPLE STAGE MODULES
You are using modules with 3 stages.
The number of thermocouples in stage 1 is 127
The number of thermocouples in stage 2 is 71

The number of thermocouples in stage 3 is 31

Tambient cold = 71.60, Tambient hot = 115.00 [deg F]
Rcold = .0527, Rhot = .0527 [deg F-hr/Btu]
COPhot = COPcold + 1, EERhot = EERcold + 3.412
Desired cooling load is 68246. [Btu/hr]
Desired heating load is 34123. [Btu/hr]

Current	Module Temp.		Q per module		COP	EER	Voltage	Number of	
[amps]	[deg F]		[Btu/hr]				[volts]	modules	
cool	heat	cool	heat	cool	cool		cool	heat	
3.0	71.31	116.44	5.50	27.30	.253	.862	2.129	12400	1251
3.5	70.83	117.28	14.57	43.18	.510	1.739	2.395	4683	791
4.0	70.37	118.15	23.42	59.73	.645	2.201	2.660	2915	572
4.5	69.91	119.06	32.04	76.95	.713	2.435	2.925	2130	444
5.0	69.47	120.00	40.45	94.85	.744	2.537	3.188	1688	360

The geometric factor for your module is 1.2550000

HEAT PUMP DESIGN - SINGLE STAGE MODULES

Tambient cold = 71.00, Tambient hot = 95.00 [deg F]
Rcold = .0527, Rhot = .0527 [deg F-hr/Btu]
COPhot = COPcold + 1, EERhot = EERcold + 3.412
Desired cooling load is 68250. [Btu/hr]
Desired heating load is 34125. [Btu/hr]

Current	Module Temp.		Q per module		COP	EER	Voltage	Number of	
[amps]	[deg F]		[Btu/hr]				[volts]	modules	
cool	heat	cool	heat	cool	cool		cool	heat	
5.0	70.80	95.61	3.85	11.65	.494	1.684	.457	17721	2929
10.0	68.43	98.96	48.77	75.23	1.843	6.289	.775	1400	454
15.0	66.34	102.62	88.51	144.58	1.579	5.388	1.095	772	237
20.0	64.51	106.59	123.15	219.90	1.273	4.345	1.417	555	156
25.0	62.95	110.89	152.73	301.44	1.028	3.506	1.742	447	114
30.0	61.66	115.53	177.26	389.51	.835	2.850	2.073	386	88
35.0	60.63	120.53	196.74	484.36	.684	2.334	2.408	347	71
40.0	59.87	125.90	211.11	586.36	.563	1.920	2.749	324	59
45.0	59.39	131.67	220.32	695.92	.463	1.581	3.097	310	50
50.0	59.18	137.87	224.27	813.46	.381	1.299	3.453	305	42
55.0	59.26	144.51	222.81	939.49	.311	1.061	3.818	307	37
60.0	59.63	151.63	215.79	1074.56	.251	.858	4.194	317	32
65.0	60.30	159.26	202.98	1219.29	.200	.682	4.581	337	28
70.0	61.30	167.43	184.13	1374.36	.155	.528	4.982	371	25
75.0	62.63	176.22	158.73	1541.19	.115	.392	5.402	430	23

Module manufacturer is MELCOR.
The geometric factor for your module is 1.2550000

HEAT PUMP DESIGN - MULTIPLE STAGE MODULES
You are using modules with 3 stages.
The number of thermocouples in stage 1 is 127
The number of thermocouples in stage 2 is 71
The number of thermocouples in stage 3 is 31

Tambient cold = 71.60, Tambient hot = 95.00 [deg F]
Rcold = .0527, Rhot = .0527 [deg F-hr/Btu]
COPhot = COPcold + 1, EERhot = EERcold + 3.412
Desired cooling load is 68246. [Btu/hr]
Desired heating load is 34123. [Btu/hr]

Current	Module Temp.		Q per module		COP	EER	Voltage	Number of	
[amps]	[deg F]		[Btu/hr]				[volts]	modules	
cool	heat	cool	heat	cool	cool			cool	heat
1.5	71.55	95.33	.91	6.29	.168	.574	1.052	75265	5423
2.0	71.05	96.02	10.49	19.43	1.173	4.001	1.311	6507	1756
2.5	70.55	96.75	19.84	33.21	1.483	5.061	1.568	3441	1028
3.0	70.07	97.51	28.96	47.63	1.551	5.293	1.824	2357	717
3.5	69.60	98.30	37.86	62.69	1.525	5.204	2.079	1803	545
4.0	69.15	99.13	46.54	78.38	1.462	4.988	2.333	1467	436
4.5	68.70	99.99	55.01	94.72	1.386	4.728	2.586	1241	361
5.0	68.27	100.89	63.27	111.70	1.307	4.458	2.838	1079	306

Module manufacturer is MELCOR.
The geometric factor for your module is 1.2550000

HEAT PUMP DESIGN - MULTIPLE STAGE MODULES

You are using modules with 3 stages.
The number of thermocouples in stage 1 is 127
The number of thermocouples in stage 2 is 71
The number of thermocouples in stage 3 is 31

Tambient cold = 71.00, Tambient hot = 115.00 [deg F]
Rcold = .0527, Rhot = .0527 [deg F-hr/Btu]
COPhot = COPcold + 1, EERhot = EERcold + 3.412
Desired cooling load is 68250. [Btu/hr]
Desired heating load is 34125. [Btu/hr]

Current	Module Temp.		Q per module		COP	EER	Voltage	Number of	
[amps]	[deg F]		[Btu/hr]				[volts]	modules	
cool	heat	cool	heat	cool	cool			cool	heat
3.0	70.75	116.40	4.68	26.54	.214	.731	2.135	14575	1286
3.5	70.28	117.24	13.74	42.42	.479	1.635	2.401	4967	805
4.0	69.81	118.11	22.58	58.97	.620	2.117	2.666	3023	579
4.5	69.36	119.02	31.19	76.19	.693	2.365	2.931	2189	448
5.0	68.91	119.96	39.59	94.09	.726	2.479	3.194	1724	363

K.5 Vapor Compression Cycle Heat Pump Programs.

K.5.1 AMMONIA.FOR

This program is the same as the one used for the complex compound heat pump, with the option selected which substitutes a mechanical compressor for the complex compound absorption/desorption. The properties of ammonia used in the program are taken from formulas given in W. C. Reynolds; *Thermodynamic Properties in SI*, Stanford University 1979.

The data to be input is the following:

- Drive to be used for output files
- Desired units
- Options:
  - Given ammonia temperature and volume, find pressure
  - Given ammonia temperature and pressure, find volume
  - Given ammonia pressure and volume, find temperature
  - Vapor-compression cycle
  - Complex compound cycle
- Evaporator temperature or pressure
- Condenser temperature or pressure
- (The following is for the vapor-compression option)
- Degrees of superheat
- Compressor efficiency
- Size on the basis of heating or cooling load

Sample output for AMMONIA.FOR - SI units

AMMONIA VAPOR-COMPRESSION & COMPLEX COMPOUND
HEAT PUMP ANALYSIS.

REFRIGERATION/HEATING CYCLE ANALYSIS FOR AMMONIA
VAPOR-COMPRESSION CYCLE

	EVAPORATOR		COMPRESSOR	CONDENSER	
Pressure [kPa]	913.797			1350.477	
	INLET	OUTLET	INLET	INLET	OUTLET
Temperature [deg C]	22.00	22.00	22.00	53.87	35.00
Volume [m^3/kg]	.0091	.1403	.1403	.1056	.001702
Enthalpy [kJ/kg]	492.71	1607.18	1607.18	1672.27	492.71
Entropy [kJ/(kg K)]	1.9604	5.7363	5.7363	5.7798	1.9534
Quality	.0535				

CYCLE PERFORMANCE		LOAD AND POWER REQUIREMENTS	
Cooling COP	17.12	Heating COP	18.12
Cooling EER	58.42	Heating EER	61.84
Cooling load [kW]	20.000	Heating load [kW]	21.168
Compressor efficiency (%)	80.0	Compressor power [kW]	1.168
Degrees superheat [deg C]	.0	Mass flow rate [kg/hr]	64.60

Sample output for AMMONIA.FOR - English units

AMMONIA VAPOR-COMPRESSION & COMPLEX COMPOUND HEAT PUMP ANALYSIS.

REFRIGERATION/HEATING CYCLE ANALYSIS FOR AMMONIA VAPOR-COMPRESSION CYCLE

	EVAPORATOR		COMPRESSOR	CONDENSER	
Pressure [psia]	132.535			195.870	
	INLET	OUTLET	INLET	INLET	OUTLET
Temperature [deg F]	71.60	71.60	71.60	128.97	95.00
Volume [ft^3/lbm]	.1451	2.2473	2.2473	1.6908	.027267
Enthalpy [Btu/lbm]	211.84	691.01	691.01	718.99	211.84
Entropy [Btu/(lbm R)	.4682	1.3701	1.3701	1.3805	.4666
Quality	.0535				

CYCLE PERFORMANCE		LOAD AND POWER REQUIREMENTS	
Cooling COP	17.12	Heating COP	18.12
Cooling EER	58.42	Heating EER	61.84
Cooling load [Btu/hr]	68246.000	Heating load [Btu/hr]	72231.840
Compressor efficiency (%)	80.0	Compressor power [kW]	1.168
Degrees superheat [deg F]	.0	Mass flow rate [lbm/hr]	142.43

K.5.2 R-134a.FOR

This program is essentially the same as AMMONIA.FOR, with the properties of refrigerant R-134a substituted for that of ammonia. The properties of R-134a used in the program are taken from formulas given by R. Tillner-Roth and H. D. Baehr, *An international standard formulation of the thermodynamic properties of 1,1,1,2-tetrafluoroethane (HFC-134a) covering temperatures from 170 K to 455 K at pressures up to 70 MPa*, a manuscript which is scheduled to appear shortly in the published literature.

The data to be input is the following:

- Drive to be used for output files
- Desired units
- Select option:
  - Vapor properties: Given T, V, find p; Given T, p, find V; given p, V, find T
  - Liquid properties: Given T, V, find p; Given T, p, find V; given p, V, find T
  - Saturation properties: Given T find p; Given p find T
  - Vapor compression cycle
- (The following is for the vapor compression cycle)
- Either evaporator pressure or temperature may be specified
- Degrees of superheat at the compressor inlet
- Compressor efficiency

Either condenser pressure or temperature may be specified
Cooling or heating load

Sample output for R134a.FOR - SI units

REFRIGERATION/HEATING CYCLE ANALYSIS FOR R-134a
VAPOR-COMPRESSION CYCLE

	EVAPORATOR		COMPRESSOR	CONDENSER	
Pressure [kPa]	607.896			886.976	
	INLET	OUTLET	INLET	INLET	OUTLET
Temperature [deg C]	22.00	22.00	22.00	38.32	35.00
Volume [m^3/kg]	.0043	.0339	.0339	.0236	.000857
Enthalpy [kJ/kg]	250.40	411.93	411.93	421.80	250.40
Entropy [kJ/(kg K)]	1.1739	1.7212	1.7212	1.7286	1.1715
Quality	.1052				

CYCLE PERFORMANCE		LOAD AND POWER REQUIREMENTS	
Cooling COP	16.37	Cooling load [kW]	20.000
Heating COP	17.37	Heating load [kW]	21.222
Compressor efficiency (%)	80.0	Compressor power [kW]	1.222
Degrees superheat [deg C]	.0	Mass flow rate [kg/hr]	445.726
Cooling EER	55.85		
Heating EER	59.26		

Sample output for R134a.FOR - English units

REFRIGERATION/HEATING CYCLE ANALYSIS FOR R-134a
VAPOR-COMPRESSION CYCLE

	EVAPORATOR		COMPRESSOR	CONDENSER	
Pressure [psia]	88.168			128.645	
	INLET	OUTLET	INLET	INLET	OUTLET
Temperature [deg F]	71.60	71.60	71.60	100.98	95.01
Volume [ft^3/lbm]	.0688	.5423	.5423	.3773	.013720
Enthalpy [Btu/lbm]	107.65	177.10	177.10	181.34	107.65
Entropy [Btu/(lbm R)]	.2804	.4111	.4111	.4129	.2798
Quality	.1052				

CYCLE PERFORMANCE		LOAD AND POWER REQUIREMENTS	
Cooling COP	16.37	Cooling load [Btu/hr]	68246.000
Heating COP	17.37	Heating load [Btu/hr]	72415.520
Compressor efficiency (%)	80.0	Compressor power [kW]	1.222
Degrees superheat [deg F]	.0	Mass flow rate [lbm/hr]	982.658
Cooling EER	55.85		
Heating EER	59.26		

K.5.3 BRAYTON.FOR

The data to be input is the following:

- Drive to be used for output files
- Desired units
- Ambient temperature
- Cooling or heating
- Cooling or heating load
- Desired room temperature
- Compressor and turbine efficiency
- Air entering from outside or inside
- Type of analysis
  - Ideal (no thermal resistance)
    - Pressure ratio or compute optimum
  - Real analysis (thermal resistance included)
    - Thermal resistance of heat exchanger

Sample output for BRAYTON.FOR - SI units

MODIFIED BRAYTON CYCLE (open cycle, gas is air)
IDEAL ANALYSIS COOLING - thermal resistance excluded

Room Temperature = 22.0 [deg C]
Cooling Load = 20.0 [kW]
Compressor efficiency = 80.0 %
Turbine efficiency = 80.0 %
Outside air enters the compressor.

	Temperatures [deg C]						
Pressure		Compressor		Turbine			
Ratio	Ambient	Entering	Exiting	Entering	Exiting	COPcool	EERcool
1.936	35.0	35.0	115.0	35.0	-7.4	.782	2.667
1.999	36.0	36.0	120.5	36.0	-8.4	.757	2.583
2.061	37.0	37.0	126.0	37.0	-9.3	.734	2.505
2.125	38.0	38.0	131.5	38.0	-10.2	.712	2.431
2.189	39.0	39.0	136.9	39.0	-11.1	.692	2.361
2.254	40.0	40.0	142.3	40.0	-11.9	.673	2.296
2.320	41.0	41.0	147.8	41.0	-12.7	.655	2.234
2.387	42.0	42.0	153.2	42.0	-13.5	.637	2.175
2.455	43.0	43.0	158.6	43.0	-14.2	.621	2.119
2.524	44.0	44.0	164.0	44.0	-15.0	.605	2.066
2.593	45.0	45.0	169.5	45.0	-15.7	.590	2.015
2.664	46.0	46.0	174.9	46.0	-16.3	.576	1.966

REAL ANALYSIS COOLING - thermal resistance included

Room Temperature = 22.0 [deg C]
Cooling Load = 20.0 [kW]
Compressor efficiency = 80.0 %
Turbine efficiency = 80.0 %
Outside air enters the compressor.
Thermal Resistance = .50000 [deg C/kW]

Pressure		Temperatures [deg C]					
		Compressor		Turbine			
Ratio	Ambient	Entering	Exiting	Entering	Exiting	COPcool	EERcool
2.612	35.0	35.0	156.6	79.7	12.0	.186	.634
2.749	36.0	36.0	165.5	83.6	12.0	.173	.590
2.898	37.0	37.0	174.7	87.7	12.0	.161	.550
3.060	38.0	38.0	184.4	91.9	12.0	.150	.513
3.236	39.0	39.0	194.5	96.2	12.0	.140	.478
3.428	40.0	40.0	205.1	100.8	12.0	.131	.447
3.639	41.0	41.0	216.3	105.5	12.0	.122	.417
3.873	42.0	42.0	228.1	110.5	12.0	.114	.389
4.133	43.0	43.0	240.6	115.7	12.0	.106	.363
4.425	44.0	44.0	253.9	121.2	12.0	.099	.339
4.755	45.0	45.0	268.2	127.1	12.0	.093	.316
5.133	46.0	46.0	283.7	133.4	12.0	.086	.294

Sample output for BRAYTON.FOR - English units.

MODIFIED BRAYTON CYCLE (open cycle, gas is air)

IDEAL ANALYSIS COOLING - thermal resistance excluded

Room Temperature = 71.6 [deg F]
Cooling Load = 68246.0 [Btu/hr]
Compressor efficiency = 80.0 %
Turbine efficiency = 80.0 %
Outside air enters the compressor.

Pressure		Temperatures [deg F]					
		Compressor		Turbine			
Ratio	Ambient	Entering	Exiting	Entering	Exiting	COPcool	EERcool
1.936	95.0	95.0	239.1	95.0	18.7	.782	2.667
1.971	96.0	96.0	244.6	96.0	17.7	.768	2.620
2.005	97.0	97.0	250.1	97.0	16.7	.754	2.574
2.040	98.0	98.0	255.5	98.0	15.8	.742	2.530
2.075	99.0	99.0	261.0	99.0	14.8	.729	2.488
2.111	100.0	100.0	266.5	100.0	13.9	.717	2.447
2.146	101.0	101.0	271.9	101.0	13.1	.706	2.407
2.182	102.0	102.0	277.3	102.0	12.2	.694	2.369
2.218	103.0	103.0	282.8	103.0	11.4	.683	2.332
2.254	104.0	104.0	288.2	104.0	10.5	.673	2.296
2.291	105.0	105.0	293.6	105.0	9.7	.663	2.261
2.328	106.0	106.0	299.0	106.0	8.9	.653	2.227
2.365	107.0	107.0	304.5	107.0	8.2	.643	2.194
2.402	108.0	108.0	309.9	108.0	7.4	.634	2.162
2.440	109.0	109.0	315.3	109.0	6.7	.625	2.131
2.478	110.0	110.0	320.7	110.0	5.9	.616	2.101
2.516	111.0	111.0	326.2	111.0	5.2	.607	2.071
2.555	112.0	112.0	331.6	112.0	4.5	.599	2.043
2.593	113.0	113.0	337.0	113.0	3.8	.590	2.015
2.633	114.0	114.0	342.5	114.0	3.1	.583	1.988
2.672	115.0	115.0	347.9	115.0	2.4	.575	1.961

REAL ANALYSIS COOLING - thermal resistance included

Room Temperature = 71.6 [deg F]
Cooling Load = 68246.0 [Btu/hr]
Compressor efficiency = 80.0 %
Turbine efficiency = 80.0 %

Outside air enters the compressor.
Thermal Resistance = .00026 [deg F-hr/Btu]

Pressure Ratio	Temperatures [deg F] Ambient	Compressor Entering	Compressor Exiting	Turbine Entering	Turbine Exiting	COPcool	EERcool
2.592	95.0	95.0	311.8	174.8	53.9	.185	.631
2.666	96.0	96.0	320.6	178.6	53.9	.178	.607
2.743	97.0	97.0	329.5	182.5	53.9	.171	.583
2.824	98.0	98.0	338.7	186.6	53.9	.164	.561
2.909	99.0	99.0	348.1	190.6	53.9	.158	.539
2.997	100.0	100.0	357.7	194.8	53.9	.152	.519
3.090	101.0	101.0	367.5	199.1	53.9	.146	.499
3.187	102.0	102.0	377.6	203.4	53.9	.141	.480
3.289	103.0	103.0	388.0	207.8	53.9	.135	.462
3.397	104.0	104.0	398.6	212.4	53.9	.130	.445
3.510	105.0	105.0	409.6	217.1	53.9	.125	.428
3.629	106.0	106.0	420.9	221.8	53.9	.121	.412
3.756	107.0	107.0	432.5	226.7	53.9	.116	.397
3.889	108.0	108.0	444.4	231.8	53.9	.112	.382
4.031	109.0	109.0	456.8	237.0	53.9	.108	.368
4.182	110.0	110.0	469.6	242.3	53.9	.104	.354
4.343	111.0	111.0	482.9	247.8	53.9	.100	.340
4.515	112.0	112.0	496.7	253.5	53.9	.096	.327
4.700	113.0	113.0	511.0	259.4	53.9	.092	.315
4.899	114.0	114.0	526.0	265.6	53.9	.089	.302
5.113	115.0	115.0	541.7	272.0	53.9	.085	.290